A Textbook of

INTERNAL COMBUSTION ENGINES

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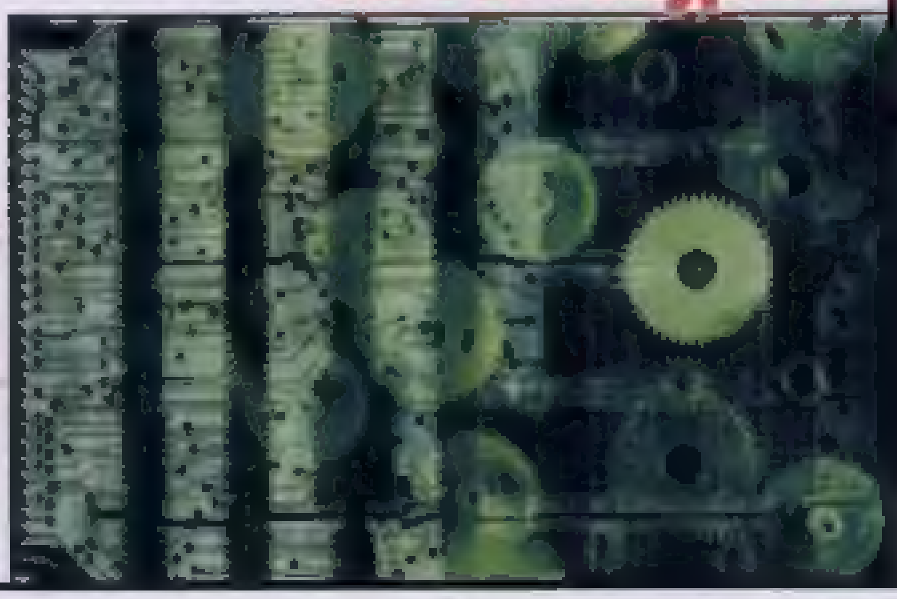

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APPENDIX **1—22**

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TO ALMIGHTY

1

Basic Concepts of Thermodynamics

1.1. Definition of thermodynamics. 1.2. Thermodynamic systems—System, boundary and surroundings—Closed system—Open system—Isolated system—Adiabatic system—Homogeneous system—Heterogeneous system. 1.3. Pure substance. 1.4. Thermodynamic equilibrium. 1.5. Properties of systems. 1.6. State. 1.7. Process. 1.8. Cycle. 1.9. Point function. 1.10. Path function. 1.11. Temperature. 1.12. Zeroth law of thermodynamics. 1.13. Pressure—Definition of pressure—Unit for pressure—Types of pressure measurement devices. 1.14. Reversible and irreversible processes. 1.15. Energy, work and heat—Energy—Work and heat. 1.16. First law of thermodynamics. 1.17. The perfect gas—The characteristic equation of state—Specific heats—Joule's law—Relationship between two specific heats—Enthalpy—Ratio of specific heats. 1.18. Steady flow energy equation (S.F.E.E.)—Energy relations for flow process. 1.19. Limitations of first law of thermodynamics. 1.20. Performance of heat engine and reversed heat engine. 1.21. Statements of second law of thermodynamics—Clausius statement. Kelvin-Planck statement. 1.22. Entropy—Introduction—Temperature entropy diagram—Characteristics of entropy. 1.23. The third law of thermodynamics. 1.24. Available and unavailable energy—Highlights—Objective Type Questions—Theoretical Questions.

1.1. DEFINITION OF THERMODYNAMICS

Thermodynamics may be defined as follows :

Thermodynamics is an axiomatic science which deals with the relations among heat, work and properties of system which are in equilibrium. It describes state and changes in state of physical systems.

Or

Thermodynamics is *the science of the regularities governing processes of energy conversion.*

Or

Thermodynamics is *the science that deals with the interaction between energy and material systems.*

Thermodynamics, basically entails *four laws* or axioms known as Zeroth, First, Second and Third law of thermodynamics.

— the *First law* throws light on *concept of internal energy*.

— the *Zeroth law* deals with *thermal equilibrium* and establishes a *concept of temperature*.

— the *Second law* indicates the limit of *converting heat into work* and introduces the *principle of increase of entropy*.

— third law defines the *absolute zero of entropy*.

These laws are based on experimental observations and have no *mathematical proof*. Like all physical laws, these laws are based on *logical reasoning*.

1.2. THERMODYNAMIC SYSTEMS

1.2.1. System, Boundary and Surroundings

System. A system is *a finite quantity of matter or a prescribed region of space* (Refer Fig. 1.1)

Boundary. The *actual or hypothetical envelope enclosing the system* is the boundary of the system. The boundary may be fixed or it may move, as and when a system containing a gas is compressed or expanded. The boundary may be *real* or *imaginary*. It is not difficult to envisage a real boundary but an example of imaginary boundary would be one drawn around a system consisting of the fresh mixture about to enter the cylinder of an I.C. engine together with the remanants of the last cylinder charge after the exhaust process (Refer Fig. 1.2).

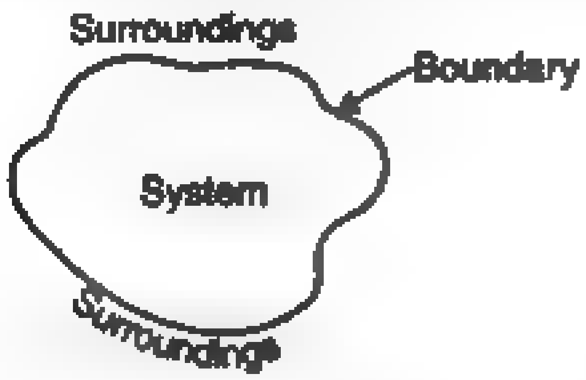


Fig. 1.1. The system.

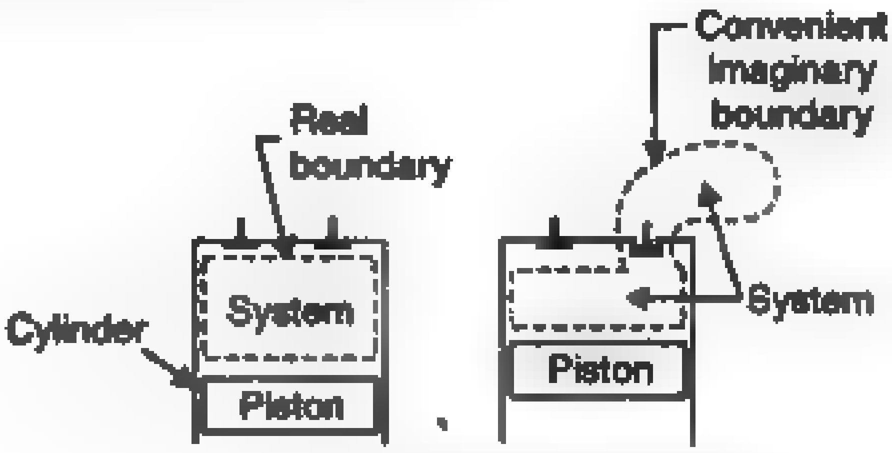


Fig. 1.2. The real and imaginary boundaries.

1.2.2. Closed System

Refer Fig. 1.3. If the boundary of the system is impervious to the flow of matter, it is called a *closed system*. An example of this system is mass of gas or vapour contained in an engine cylinder, the boundary of which is drawn by the cylinder walls, the cylinder head and piston crown. Here the *boundary is continuous and no matter may enter or leave*.

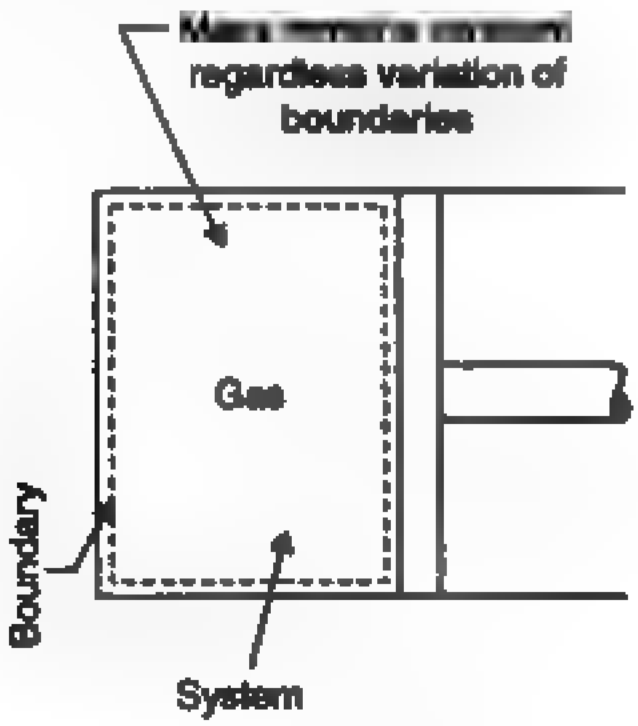


Fig. 1.3. Closed system.

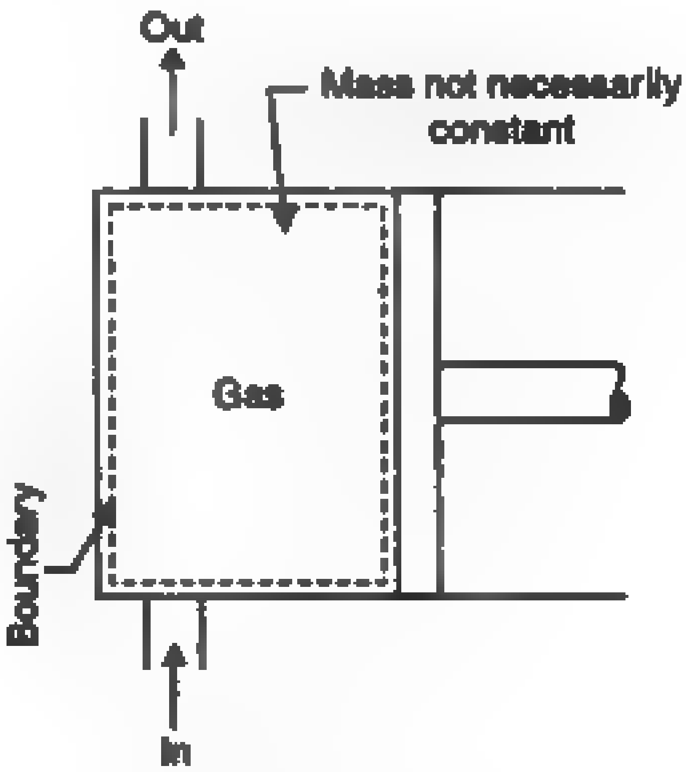


Fig. 1.4. Open system.

1.2.3. Open System

Refer Fig. 1.4. An open system is one in which *matter flows into or out of the system*. Most of the engineering systems are open.

1.2.4. Isolated System

An isolated system is that system *which exchanges neither energy nor matter with any other system or with environment.*

1.2.5. Adiabatic System

An adiabatic system is one *which is thermally insulated from its surroundings.* It can, however, *exchange work with its surroundings.* If it does not, it becomes an isolated system.

Phase. A phase is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.

1.2.6. Homogeneous System

A system which consists of a single phase is termed as *homogeneous system.* Examples : Mixture of air and water vapour, water plus nitric acid and octane plus heptane.

1.2.7. Heterogeneous System

A system which consists of two or more phases is called a *heterogeneous system.* Examples : Water plus steam, ice plus water and water plus oil.

1.3. PURE SUBSTANCE

A pure substance is one that has a homogeneous and invariable chemical composition even though there is a change of phase. In other words, it is a system which is (*a*) homogeneous in composition, (*b*) homogeneous in chemical aggregation. Examples : Liquid, water, mixture of liquid water and steam, mixture of ice and water. The mixture of liquid air and gaseous air is not a pure substance.

1.4. THERMODYNAMIC EQUILIBRIUM

A system is in *thermodynamic equilibrium* if the temperature and pressure at all points are same ; there should be no velocity gradient ; the chemical equilibrium is also necessary. Systems under temperature and pressure equilibrium but not under chemical equilibrium are sometimes said to be in metastable equilibrium conditions. *It is only under thermodynamic equilibrium conditions that the properties of a system can be fixed.*

Thus for attaining a state of *thermodynamic equilibrium* the following three types of equilibrium states must be achieved :

1. **Thermal equilibrium.** The temperature of the system does not change with time and has same value at all points of the system.

2. **Mechanical equilibrium.** There are no unbalanced forces within the system or between the surroundings. The pressure in the system is same at all points and does not change with respect to time.

3. **Chemical equilibrium.** No chemical reaction takes place in the system and the chemical composition which is same throughout the system does not vary with time.

1.5. PROPERTIES OF SYSTEMS

A property of a system is a characteristic of the system which depends upon its state, but not upon how the state is reached. There are two sorts of property :

1. **Intensive properties.** These properties *do not depend on the mass of the system. Examples* : Temperature and pressure.

2. Extensive properties. These properties *depend on the mass of the system. Example* : Volume. Extensive properties are often divided by mass associated with them to obtain the intensive properties. For example, if the volume of a system of mass m is V, then the specific volume of matter within the system is $\frac{V}{m} = v$ which is an intensive property.

1.6. STATE

***State** is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a state.*

It follows from the definition of state that each property has a single value at each state. Stated differently, all properties are *state* or *point functions*. Therefore, all properties are identical for identical states.

On the basis of the above discussion, we can determine if a given variable is *property* or not by applying the following *tests* :

— *A variable is a property, if and only if, it has a single value at each equilibrium state.*

— *A variable is a property, if and only if, the change in its value between any two prescribed equilibrium states is single-valued.*

Therefore, *any variable whose change is fixed by the end states is a property.*

1.7. PROCESS

A process occurs when the system undergoes a change in a state or an energy transfer at a steady state. A process may be *non-flow* in which a fixed mass within the defined boundary is undergoing a change of state. Example : a substance which is being heated in a closed cylinder undergoes a non-flow process (Fig. 1.3). *Closed systems undergo non-flow processes.* A process may be a flow process in which mass is entering and leaving through the boundary of an open system. In a steady flow process (Fig. 1.4) mass is crossing the boundary from surroundings at entry, and an equal mass is crossing the boundary at the exit so that the total mass of the system remains constant. In an open system it is necessary to take account of the work delivered from the surroundings to the system at entry to cause the mass to enter, and also of the work delivered from the system at surroundings to cause the mass to leave, as well as any heat or work crossing the boundary of the system.

Quasi-static process. Quasi means 'almost'. A quasi-static process is also called a *reversible process. This process is a succession of equilibrium states and infinite slowness is its characteristic feature.*

1.8. CYCLE

Any process or series of processes whose end states are identical is termed a **cycle.** The processes through which the system has passed can be shown on a state diagram, but a complete section of the path requires in addition a statement of the heat and work crossing the boundary of the system. Fig. 1.5 shows such a cycle in which a system commencing at condition '1' changes in pressure and volume through a path 123 and returns to its initial condition '1'.

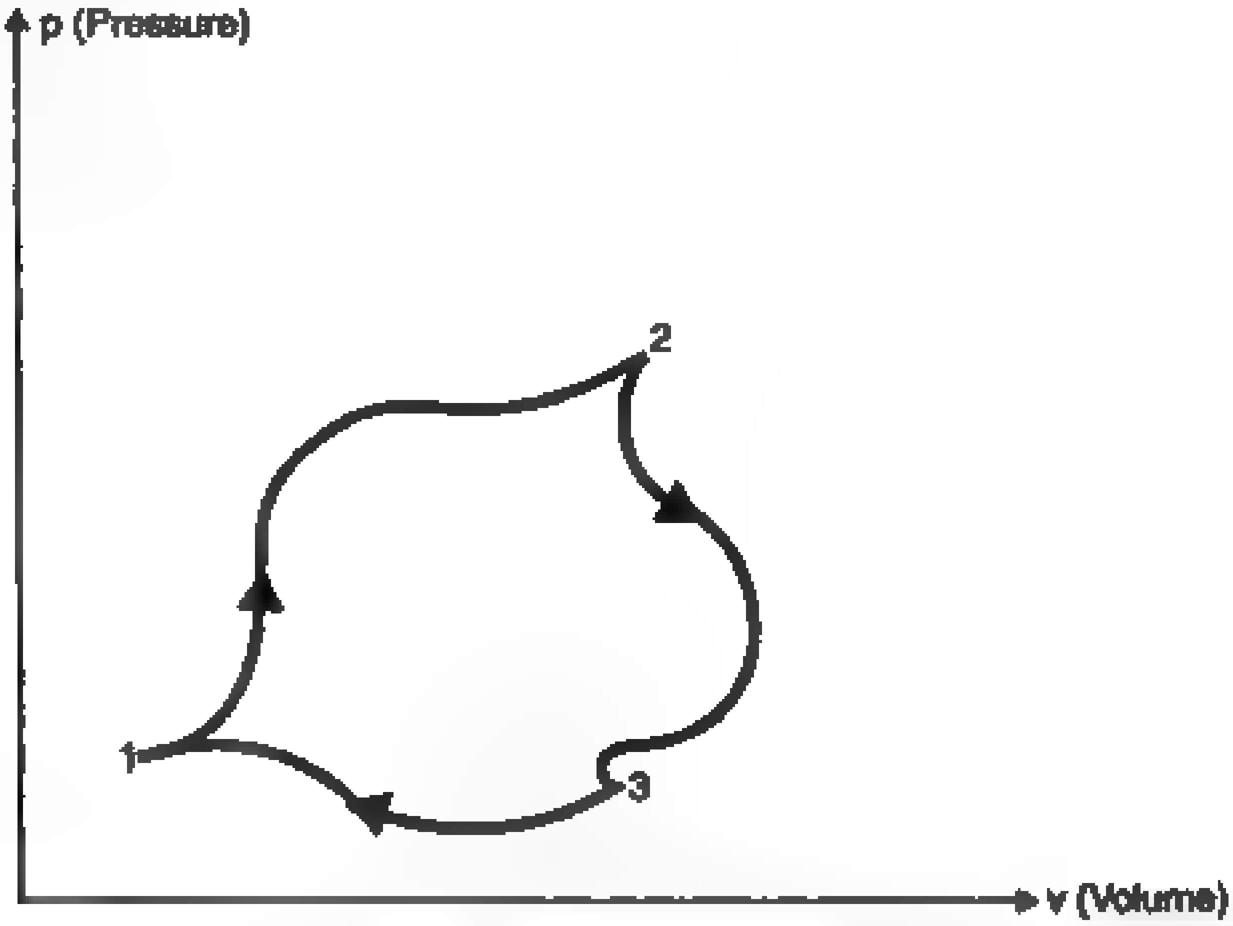


Fig. 1.5. Cycle of operations.

1.9. POINT FUNCTION

When two properties locate a *point* on the graph (co-ordinate axes) then those properties are called as *point function.*

Examples. Pressure, temperature, volume etc.

$$\int_1^2 dV = V_2 - V_1 \text{ (an \textit{exact differential})}.$$

1.10. PATH FUNCTION

There are certain quantities which cannot be located on a graph by a *point* but are given by the *area* or so, on that graph. In that case, the area on the graph, pertaining to the particular process, is a function of the path of the process. Such quantities are called *path functions.*

Examples. Heat, work etc.

Heat and work are *inexact differentials*. Their change cannot be written as difference between their end states.

Thus $\int_1^2 \delta Q \neq Q_2 - Q_1$ and is shown as ${}_1Q_2$ or Q_{1-2}

Similarly $\int_1^2 \delta W \neq W_2 - W_1$, and is shown as ${}_1W_2$ or W_{1-2}

Note. The operator δ is used to denote inexact differentials and operator d is used to denote exact differentials.

1.11. TEMPERATURE

- *The temperature is a thermal state of a body which distinguishes a hot body from a cold body.* The temperature of a body is *proportional to the stored molecular energy i.e.* the average molecular kinetic energy of the molecules in a system. (A particular molecule does not have a temperature, it has energy. The gas as a system has temperature).

- Instruments for measuring ordinary temperatures are known as "*thermometers*" and those for measuring high temperatures are known as "*pyrometers*".
- It has been found that a gas will not occupy any volume at a certain temperature. This temperature is known as *absolute zero temperature*. The temperatures measured with absolute zero as basis are called *absolute temperatures*. Absolute temperature is stated in degrees centigrade. The point of absolute temperature is found to occur at 273.15°C below the freezing point of water.

Then : Absolute temperature = Thermometer reading in °C + 273.15.

Absolute temperature in degree centigrade is known as degrees kelvin, denoted by K (SI unit).

1.12. ZEROTH LAW OF THERMODYNAMICS

- **'Zeroth law of thermodynamics'** *states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.*

Example. Refer Fig. 1.6. System '1' may consist of a mass of gas enclosed in a rigid vessel fitted with a pressure gauge. If there is no change of pressure when this system is brought into contact with system '2' a block of iron, then the two systems are equal in temperature (assuming that the systems 1 and 2 do not react each other chemically or electrically). Experiment reveals that if system '1' is brought into contact with a third system '3' again with no change of properties then systems '2' and '3' will show no change in their properties when brought into contact provided they do not react with each other chemically or electrically. Therefore, '2' and '3' must be in equilibrium.

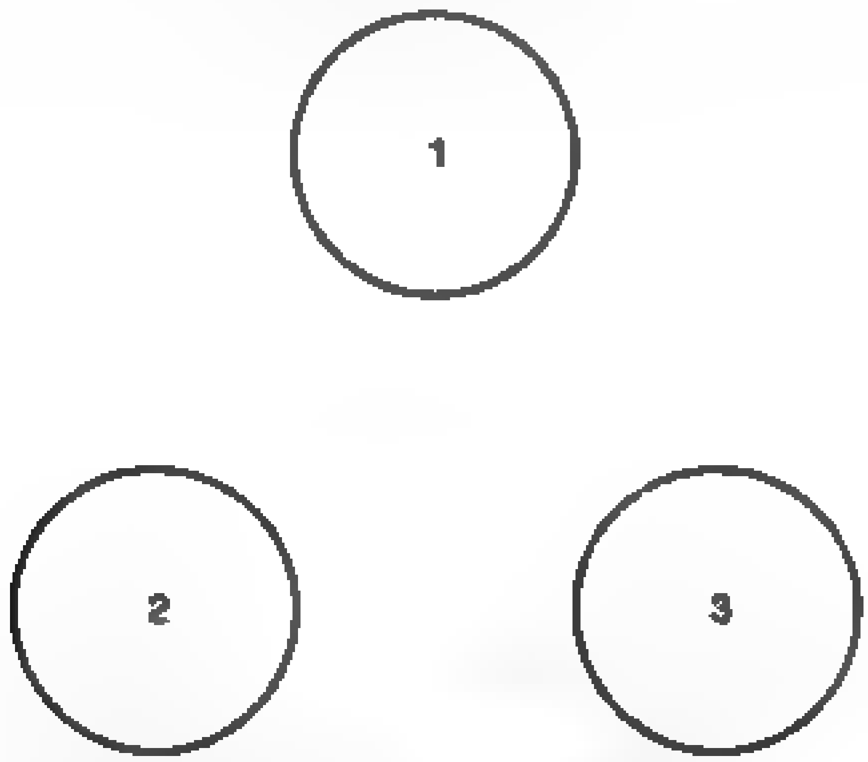


Fig. 1.6. Zeroth law of thermodynamics.

- This law was enunciated by R.H. Fowler in the year 1931. However, since the first and second laws already existed at that time, it was designated as *zeroth law* so that it *precedes* the first and second laws *to form a logical sequence.*

1.13. PRESSURE

1.13.1. Definition of Pressure

Pressure is defined as a *force per unit area*. Pressures are exerted by gases, vapours and liquids. The instruments that we generally use, however, record pressure as the difference

between two pressures. Thus, it is the *difference between the pressure exerted by a fluid of interest and the ambient atmospheric pressure.* Such devices indicate the pressure either above or below that of the atmosphere. When it is *above the atmospheric pressure*, it is termed *gauge pressure* and is *positive*. When it is *below* atmospheric, it is *negative* and is known as *vacuum*. Vacuum readings are given in millimetres of mercury or millimetres of water below the atmosphere.

It is necessary to establish an absolute pressure scale which is independent of the changes in atmospheric pressure. A pressure of absolute zero can exist only in complete vacuum. *Any pressure measured above the absolute zero of pressure* is termed an *'absolute pressure'*.

A schematic diagram showing the *gauge pressure, vacuum pressure* and the *absolute pressure* is given in Fig. 1.7.

Mathematically :

(*i*) Absolute pressure = Atmospheric pressure + Gauge pressure

$$p_{abs.} = p_{atm.} + p_{gauge}.$$

(*ii*) Vacuum pressure = Atmospheric pressure – Absolute pressure.

Vacuum is defined as the *absence of pressure*. A *perfect vacuum* is obtained when *absolute pressure is zero*, at this instant *molecular momentum is zero*.

Atmospheric pressure is measured with the help of barometer.

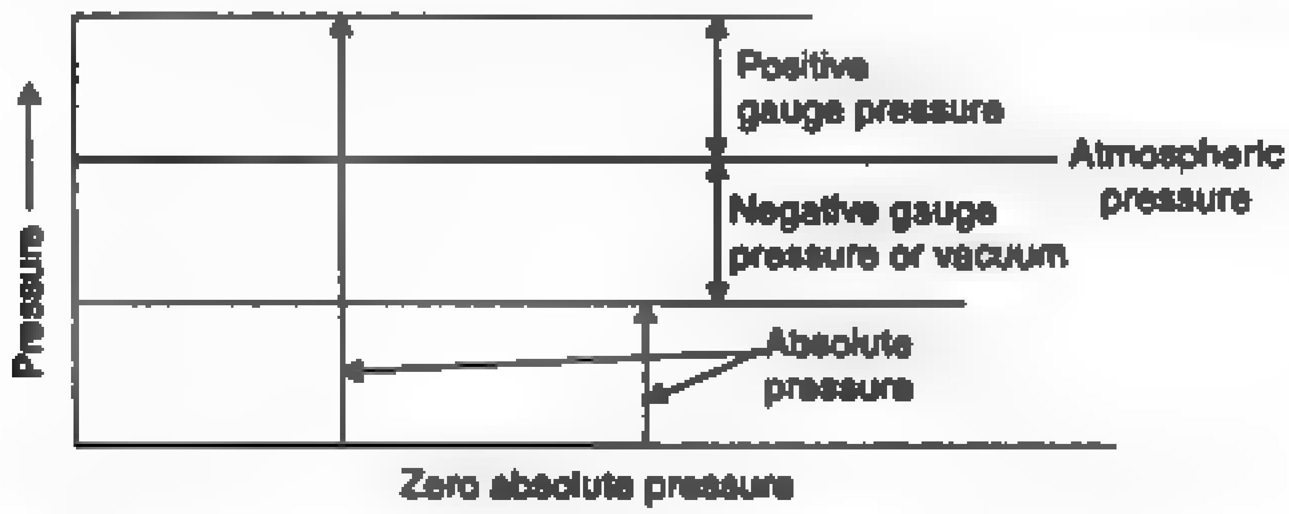


Fig. 1.7. Schematic diagram showing gauge, vacuum and absolute pressures.

1.13.2. Unit for Pressure

The fundamental SI unit of pressure is N/m^2 (sometimes called *pascal*, Pa) or bar. 1 bar = 10^5 N/m^2 = 10^5 Pa. Standard atmospheric pressure = 1.01325 bar = 0.76 m Hg.

Low pressures are often expressed in terms of mm of water or mm of mercury. This is an abbreviated way of saying that the pressure is such that which will support a liquid column of stated height.

1.13.3. Types of Pressure Measurement Devices

The pressure may be measured by means of indicating gauges or recorders. These instruments may be mechanical, electro-mechanical, electrical or electronic in operation.

1. **Mechanical instruments.** These instruments may be classified into following two groups :

— The *first group* includes those instruments in which the *pressure* measurement is made by *balancing an unknown force with a known force.*

— The *second group* includes those employing *quantitative deformation of an elastic member for pressure measurement.*

2. **Electro-mechanical instruments.** These instruments usually *employ a mechanical means for detecting the pressure and electrical means for indicating or recording the detected pressure.*

3. **Electronic instruments.** Electronic pressure measuring instruments normally depend on some physical change that can be detected and indicated or recorded electronically.

1.14. REVERSIBLE AND IRREVERSIBLE PROCESSES

Reversible process. *A reversible process (also sometimes known as quasi-static process) is one which can be stopped at any stage and reversed so that the system and surroundings are exactly restored to their initial states.*

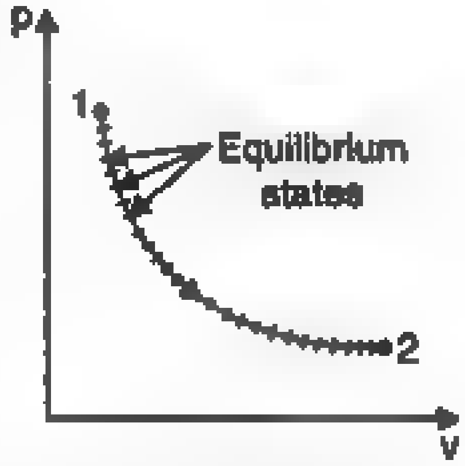


Fig. 1.8. Reversible process.

This process has the following *characteristics* :

1. It must pass through the same states on the reversed path as were initially visited on the forward path.

2. This process when undone will leave no history of events in the surroundings.

3. It must pass through a continuous series of equilibrium states.

No real process is truely reversible but some processes may approach reversibility, to close approximation.

Examples. Some *examples of nearly reversible processes* are :

(*i*) Frictionless relative motion.
(*ii*) Expansion and compression of spring.
(*iii*) Frictionless adiabatic expansion or compression of fluid.
(*iv*) Polytropic expansion or compression of fluid.
(*v*) Isothermal expansion or compression.
(*vi*) Electrolysis.

Irreversible process. An *irreversible process is one in which heat is transferred through a finite temperature.*

Examples :

(*i*) Relative motion with friction
(*ii*) Combustion
(*iii*) Diffusion
(*iv*) Free expansion
(*v*) Throttling
(*vi*) Electricity flow through a resistance
(*vii*) Heat transfer
(*viii*) Plastic deformation.

An irreversible process is usually represented by a dotted (or discontinuous) line joining the end states to indicate that the intermediate states are indeterminate (Fig. 1.9).

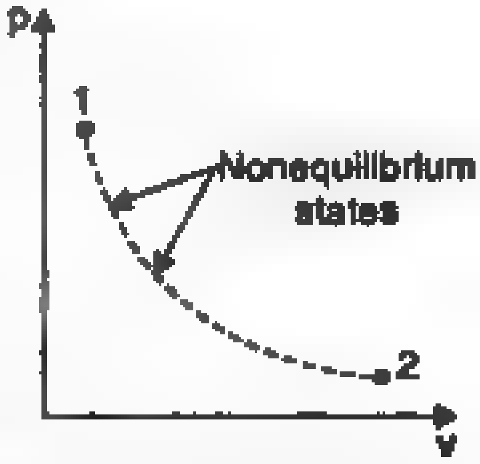


Fig. 1.9. Irreversible process.

Irreversibilities are of *two types* :

1. **External irreversibilities.** These are associated with *dissipating effects outside the working fluid.*

Example. *Mechanical friction occurring during a process due to some external source.*

2. **Internal irreversibilities.** These are associated with *dissipating effects within the working fluid.*

Example. *Unrestricted expansion of gas, viscosity and inertia of the gas.*

1.15. ENERGY, WORK AND HEAT

1.15.1. Energy

Energy is a general term embracing ***energy in transition*** and ***stored energy***. The stored energy of a substance may be in the forms of *mechanical energy* and *internal energy* (other forms of stored energy may be chemical energy and electrical energy). Part of the stored energy may take the form of either potential energy (which is the gravitational energy due to height above a chosen datum line) or kinetic energy due to velocity. The balance part of the energy is known as *internal energy*. In a ***non-flow process*** usually there is no change of potential or kinetic energy and hence change of mechanical energy will not enter the calculations. In a *flow process*, however, there may be changes in both potential and kinetic energy and these must be taken into account while considering the changes of stored energy. ***Heat and work*** are *the forms of energy in transition*. These are the only forms in which energy can cross the boundaries of a system. *Neither heat nor work can exist as stored energy.*

1.15.2. Work and Heat

Work

Work is said to be done when a *force moves through a distance*. If a part of the boundary of a system undergoes a displacement under the action of a pressure, the work done W is the product of the force (pressure × area), and the distance it moves in the direction of the force. Fig. 1.10 (*a*) illustrates this with the conventional piston and cylinder arrangement, the heavy line defining the boundary of the system. Fig. 1.10 (*b*) illustrates another way in which work might be applied to a system. A force is exerted by the paddle as it changes the momentum of the fluid, and since this force moves during rotation of the paddle room work is done.

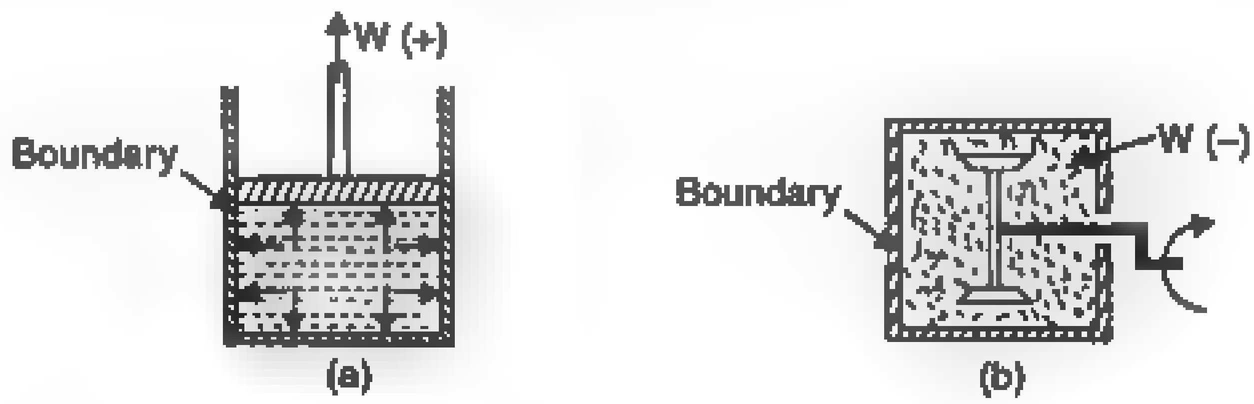


Fig. 1.10

"Work" is a transient quantity which only appears at the boundary while a change of state is taking place within a system. "Work" is 'something' which appears at the boundary when a system changes its state due to the movement of a part of the boundary under the action of a force.

Sign convention :

- If the work is done *by* the system *on* the surroundings, *e.g.* when a fluid expands pushing a piston outwards, the work is said to be *positive*.

i.e., *Work output of the system* $= + W$

- If the work is done *on* the system *by* the surroundings, *e.g.* when a force is applied to a rotating handle, or to a piston to compress a fluid, the work is said to be *negative*.

i.e., *Work input to system* $= - W$

Heat

Heat (denoted by the symbol Q), may be, defined in an analogous way to work as follows :

"Heat is 'something' which appears at the boundary when a system changes its state due to a difference in temperature between the system and its surroundings".

Heat, like work, is a *transient quantity* which only appears at the boundary while a change is taking place within the system.

It is apparent that neither δW or δQ are exact differentials and therefore any integration of the elemental quantities of work or heat which appear during a change from state 1 to state 2 must be written as

$$\int_1^2 \delta W = W_{1-2} \text{ or } {}_1W_2 \text{ (or } W\text{), and}$$

$$\int_1^2 \delta Q = Q_{1-2} \text{ or } {}_1Q_2 \text{ (or } Q\text{)}$$

Sign convention :

If the heat flows *into* a system *from* the surroundings, the quantity is said to be *positive* and, conversely, if heat flows *from* the system to the surroundings it is said to be *negative*.

In other words :

Heat received by the system $= + Q$

Heat rejected or given up by the system $= - Q$.

Comparison of Work and Heat

Similarities :

(*i*) Both are *path functions and inexact differentials*.

(*ii*) Both are boundary phenomenon *i.e.*, both are recognized at the boundaries of the system as they cross them.

(*iii*) Both are associated with a process, not a state. Unlike properties, work or heat has no meaning at a state.

(*iv*) Systems possess energy, but not work or heat.

Dissimilarities :

(*i*) In heat transfer temperature difference is required.

(*ii*) In a stable system there cannot be work transfer, however, there is no restriction for the transfer of heat.

(*iii*) The sole effect external to the system could be reduced to rise of a weight but in the case of a heat transfer other effects are also observed.

1.16. FIRST LAW OF THERMODYNAMICS

It is observed that when a system is made to undergo a complete cycle then net work is done *on* or *by* the system. Consider a cycle in which net work is done by the system. Since energy cannot be created, this mechanical energy must have been supplied from some source of energy. Now the system has been returned to its initial state : Therefore, its *intrinsic* energy is unchanged, and hence the mechanical energy has not been provided by the system itself. The only other energy involved in the cycle is the heat which was supplied and rejected in various processes. Hence, by the law of conservation of energy, the net work done by the system is equal to the net heat supplied

to the system. The First Law of Thermodynamics can, therefore, be stated as follows :

"When a system undergoes a thermodynamic cycle then the net heat supplied to the system from the surroundings is equal to net work done by the system on its surroundings.

or
$$\oint dQ = \oint dW$$

where $\oint$ represents the sum for a complete cycle.

The first law of Thermodynamics *cannot be proved analytically, but experimental evidence has repeatedly confirmed its validity*, and since no phenomenon has been shown to contradict it, the first law is accepted as a *law of nature*. It may be remarked that no restriction was imposed which limited the application of first law to reversible energy transformation. Hence the first law applies to reversible as well as irreversible transformations : For non-cyclic process, a more general formulation of first law of thermodynamics is required. A new concept which involves a term called *internal energy* fulfills this need.

— The First Law of Thermodynamics may also be stated as follows :

"Heat and work are mutually convertible but since energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant".

Or

— **"No machine can produce energy without corresponding expenditure of energy, *i.e.*, it is impossible to construct a perpetual motion machine of first kind".**

1.17. THE PERFECT GAS

1.17.1. The Characteristic Equation of State

— At temperatures that are considerably in excess of critical temperature of a fluid, and also at very low pressure, the vapour of fluid tends to obey the equation

$$\frac{pv}{T} = \text{constant} = R$$

In practice, no gas obeys this law rigidly, but many gases tend towards it.

An imaginary ideal gas which obeys this law is called a *perfect gas*, and the equation $\frac{pv}{T} = R$, is called the *characteristic equation of a state of a perfect gas*. The constant R is called the *gas constant*. Each perfect gas has a different gas constant.

Units of R are Nm/kg K or kJ/kg K.

Usually, the characteristic equation is written as

$$pv = RT \quad ...(1.1)$$

or for m kg, occupying V m^3

$$pV = mRT \quad ...(1.2)$$

— The characteristic equation in *another form*, can be derived by using kilogram-mole as a unit.

The *kilogram-mole* is defined as a quantity of a gas equivalent to M kg of the gas, where M is the molecular weight of the gas (*e.g.* since the molecular weight of oxygen is 32, then 1 kg mole of oxygen is equivalent to 32 kg of oxygen).

As per definition of the kilogram-mole, for m kg of a gas, we have

$$m = nM \qquad ...(1.3)$$

where n = Number of moles.

Note. Since the standard of mass is the kg, kilogram-mole will be written simply as mole.

Substituting for m from Eqn. (1.3) in Eqn. (1.2) gives

$$pV = nMRT$$

or

$$MR = \frac{pV}{nT}$$

According to *Avogadro's hypothesis* the volume of 1 mole of any gas is the same as the volume of 1 mole of any other gas, when the gases are at the same temperature and pressure. Therefore, $\frac{V}{n}$ is the same for all gases at the same value of p and T. That is the quantity $\frac{pV}{nT}$ is a *constant* for all gases. This constant is called *universal gas constant*, and is given the symbol, R_0.

i.e.,

$$MR = R_0 = \frac{pV}{nT}$$

or

$$pV = nR_0T \qquad ...(1.4)$$

Since $MR = R_0$, then

$$R = \frac{R_0}{M} \qquad ...(1.5)$$

It has been found experimentally that the volume of 1 mole of any prefect gas at 1 bar and 0°C is approximately 22.71 m^3.

Therefore from Eqn. (1.4),

$$R_0 = \frac{pV}{nT} = \frac{1\times 10^5 \times 22.71}{1\times 273.15}$$

$$= 8314.3 \text{ Nm/mole K}$$

Using Eqn. (1.5), the gas constant for any gas can be found when the molecular weight is known.

Example. For oxygen which has a molecular weight of 32, the gas constant

$$R = \frac{R_0}{M} = \frac{8314}{32} = 259.8 \text{ Nm/kg K}.$$

1.17.2. Specific Heats

— The specific heat of a solid or liquid is usually defined as the *heat required to raise unit mass through one degree temperature rise.*

— For small quantities, we have

$$dQ = mcdT$$

where m = Mass

c = Specific heat

dT = Temperature rise.

For a gas there are an infinite number of ways in which heat may be added between any two temperatures, and hence a *gas could have an infinite number of specific heats.* However, only two specific heats for gases are defined.

Specific heat at constant volume, c_v

and *Specific heat at constant pressure, c_p.*

We have

$dQ = m\, c_p\, dT$ For a reversible non-flow process at *constant pressure* ...(1.6)

and $dQ = m\, c_v\, dT$ For a reversible non-flow process at *constant volume* ...(1.7)

The values of c_p and c_v, for a perfect gas, are constant for any one gas at all pressures and temperatures. Hence, integrating Eqns. (1.6) and (1.7), we have

Flow of heat in a reversible constant pressure process

$$= mc_p\,(T_2 - T_1) \quad ...(1.8)$$

Flow of heat in a reversible constant volume process

$$= mc_v\,(T_2 - T_1) \quad ...(1.9)$$

In case of *real gases, c_p* and *c_v vary with temperature,* but a suitable *average value* may be used for most practical purposes.

1.17.3. Joule's Law

Joule's law states as follows :

"The internal energy of a perfect gas is a function of the absolute temperature only."

i.e., $u = f(T)$

To evaluate this function let 1 kg of a perfect gas be heated at constant volume.

According to non-flow energy equation,

$$dQ = du + dW$$

$$dW = 0, \text{ since volume remains constant}$$

$$\therefore \quad dQ = du$$

At constant volume for a perfect gas, from Eqn. (1.7), for 1 kg

$$dQ = c_v dT$$

$$\therefore \quad dQ = du = c_v dT$$

and integrating $\quad u = c_v\, T + K$, K being constant.

According to Joule's law $u = f(T)$, which means that internal energy varies linearly with absolute temperature. Internal energy can be made zero at any arbitrary reference temperature. For a perfect gas it can be assumed that $u = 0$ when $T = 0$, hence constant K is zero.

i.e. Internal energy, $u = c_v\, T$ for a perfect gas ...(1.10)

or For mass m, of a prefect gas

Internal energy, $U = mc_v\, T$ ...(1.11)

For a perfect gas, in any process between states 1 and 2, we have from Eqn. (1.11)

Gain in internal energy,

$$U_2 - U_1 = mc_v\,(T_2 - T_1) \quad ...(1.12)$$

Eqn. (1.12) gives the gains of internal energy for a perfect gas between two states *for any process, reversible* or *irreversible.*

1.17.4. Relationship Between Two Specific Heats

Consider a perfect gas being heated at constant pressure from T_1 to T_2.

According to non-flow equation,

$$Q = (U_2 - U_1) + W$$

Also for a perfect gas,

$$U_2 - U_1 = mc_v\,(T_2 - T_1)$$

$$Q = mc_v\,(T_2 - T_1) + W$$

In a constant pressure process, the work done by the fluid,

$$W = p(V_2 - V_1)$$

$$= mR(T_2 - T_1) \qquad \left[\because \begin{array}{l} p_1V_1 = mRT_1 \\ p_2V_2 = mRT_2 \\ p_1 = p_2 = p \text{ in this case} \end{array}\right]$$

On substituting $Q = mc_v\,(T_2 - T_1) + mR\,(T_2 - T_1) = m(c_v + R)\,(T_2 - T_1)$

But for a constant pressure process,

$$Q = mc_p\,(T_2 - T_1)$$

By equating the two expressions, we have

$$m(c_v + R)(T_2 - T_1) = mc_p(T_2 - T_1)$$

$\therefore$ $$c_v + R = c_p$$

or $$c_p - c_v = R \qquad ...(1.13)$$

Dividing both sides by c_v, we get

$$\frac{c_p}{c_v} - 1 = \frac{R}{c_v}$$

$\therefore$ $$c_v = \frac{R}{\gamma - 1} \qquad ...[1.13\ (a)]$$

(where $\gamma = c_p/c_v$)

Similarly, dividing both sides by c_p, we get

$$c_p = \frac{\gamma R}{\gamma - 1} \qquad ...[1.13\ (b)]$$

$$\left[\begin{array}{l} \text{In M.K.S. units} : c_p - c_v = \dfrac{R}{J};\ c_v = \dfrac{R}{J(\gamma - 1)},\ c_p = \dfrac{\gamma R}{(\gamma - 1)J} \\ \text{In SI units the value of } J \text{ is unity.} \end{array}\right]$$

1.17.5. Enthalpy

— One of the fundamental quantities which occur invariably in thermodynamics is the sum of internal energy (u) and pressure volume product (pv). This sum is called **Enthalpy** (h).

i.e., $$h = u + pv \qquad ...(1.14)$$

— The enthalpy of a fluid is the property of the fluid, since it consists of the sum of a property and the product of the two properties. Since enthalpy is a property like internal energy, pressure, specific volume and temperature, it can be introduced into any problem whether the process is a flow or a non-flow process.

The total enthalpy of mass, m, of a fluid can be

$$H = U + pV, \text{ where } H = mh.$$

For a *perfect gas*,

Referring equation (1.14), $h = u + pv$

$$= c_vT + RT \qquad [\because\ pv = RT]$$

$$= (c_v + R)T$$

$$= c_pT \qquad [\because\ c_p = c_v + R]$$

i.e., $$h = c_pT$$

and $$H = mc_pT.$$

(Note that, since it has been assumed that $u = 0$ at $T = 0$, then $h = 0$ at $T = 0$).

1.17.6. Ratio of Specific Heats

The ratio of specific heat at constant pressure to the specific heat at constant volume is given by the symbol γ (gamma).

i.e., $$\gamma = \frac{c_p}{c_v} \quad \text{...(1.15)}$$

Since $c_p = c_v + R$, it is clear that c_p must be *greater* than c_v for any perfect gas. It follows, therefore, that the ratio, $\frac{c_p}{c_v} = \gamma$ is *always greater than unity.*

In general, the approximate values of γ are as follows :

For *monoatomic* gases such as *argon, helium* = 1.6.

For *diatomic* gases such as *carbon monoxide, hydrogen, nitrogen* and *oxygen* = 1.4.

For triatomic gases such as *carbondioxide* and *sulphur dioxide* = 1.3.

For some hydro-carbons the value of γ is quite low.

[*e.g.*, for ethane γ = 1.22, and for isobutane γ = 1.11]

Table 1.1. Summary of Processes for Perfect Gas (Unit mass)

Process	*Index* n	*Heat added*	$\int_1^2 p\,dv$	*p, v, T relations*	*Specific heat, c*
Constant pressure	$n = 0$	$c_p(T_2 - T_1)$	$p(v_2 - v_1)$	$\frac{T_2}{T_1} = \frac{v_2}{v_1}$	c_p
Constant volume	$n = \infty$	$c_v(T_2 - T_1)$	0	$\frac{T_1}{T_2} = \frac{p_1}{p_2}$	c_v
Constant temperature	$n = 1$	$p_1v_1 \log_e \frac{v_2}{v_1}$	$p_1v_1 \log_e \frac{v_2}{v_1}$	$p_1v_1 = p_2v_2$	∞
Reversible adiabatic	$n = \gamma$	0	$\frac{p_1v_1 - p_2v_2}{\gamma - 1}$	$p_1v_1^{\gamma} = p_2v_2^{\gamma}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1}$ $= \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$	0
Polytropic	$n = n$	$c_n(T_2 - T_1)$ $= c_v\left(\frac{\gamma - n}{1 - n}\right) \times (T_2 - T_1)$ $= \frac{\gamma - n}{\gamma - 1} \times$ work done (non–flow)	$\frac{p_1v_1 - p_2v_2}{n - 1}$	$p_1v_1^{n} = p_2v_2^{n}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n - 1}$ $= \left(\frac{p_2}{p_1}\right)^{\frac{n - 1}{n}}$	$c_n = c_v\left(\frac{\gamma - n}{1 - n}\right)$

Note. Equations must be used keeping dimensional consistence.

1.18. STEADY FLOW ENERGY EQUATION (S.F.E.E.)

In many practical problems, the rate at which the fluid flows through a machine or piece of apparatus is constant. This type of flow is called *steady flow*.

Assumptions :

The following *assumptions* are made in the system analysis :

(*i*) The mass flow through the system remains constant.

(*ii*) Fluid is uniform in composition.

(*iii*) The only interaction between the system and surroundings are work and heat.

(*iv*) The state of fluid at any point remains constant with time.

(*v*) In the analysis only potential, kinetic and flow energies are considered.

Fig. 1.11 shows a schematic flow process for an open system. An open system is one in which both mass and energy may cross the boundaries. A wide interchange of energy may take place within an open system. Let the system be an automatic engine with the inlet manifold at the first state point and exhaust pipe as the second point. There would be an interchange of chemical energy in the fuel, kinetic energy of moving particles, internal energy of gas and heat transferred and shaft work within the system. From Fig. 1.11 it is obvious that if there is no variation of flow of mass or energy with time across the boundaries of the system the steady flow will prevail. The conditions may pass through the cyclic or non-cyclic changes within the system. As a result the mass entering the system equals the mass leaving, also energy entering the system equals energy leaving.

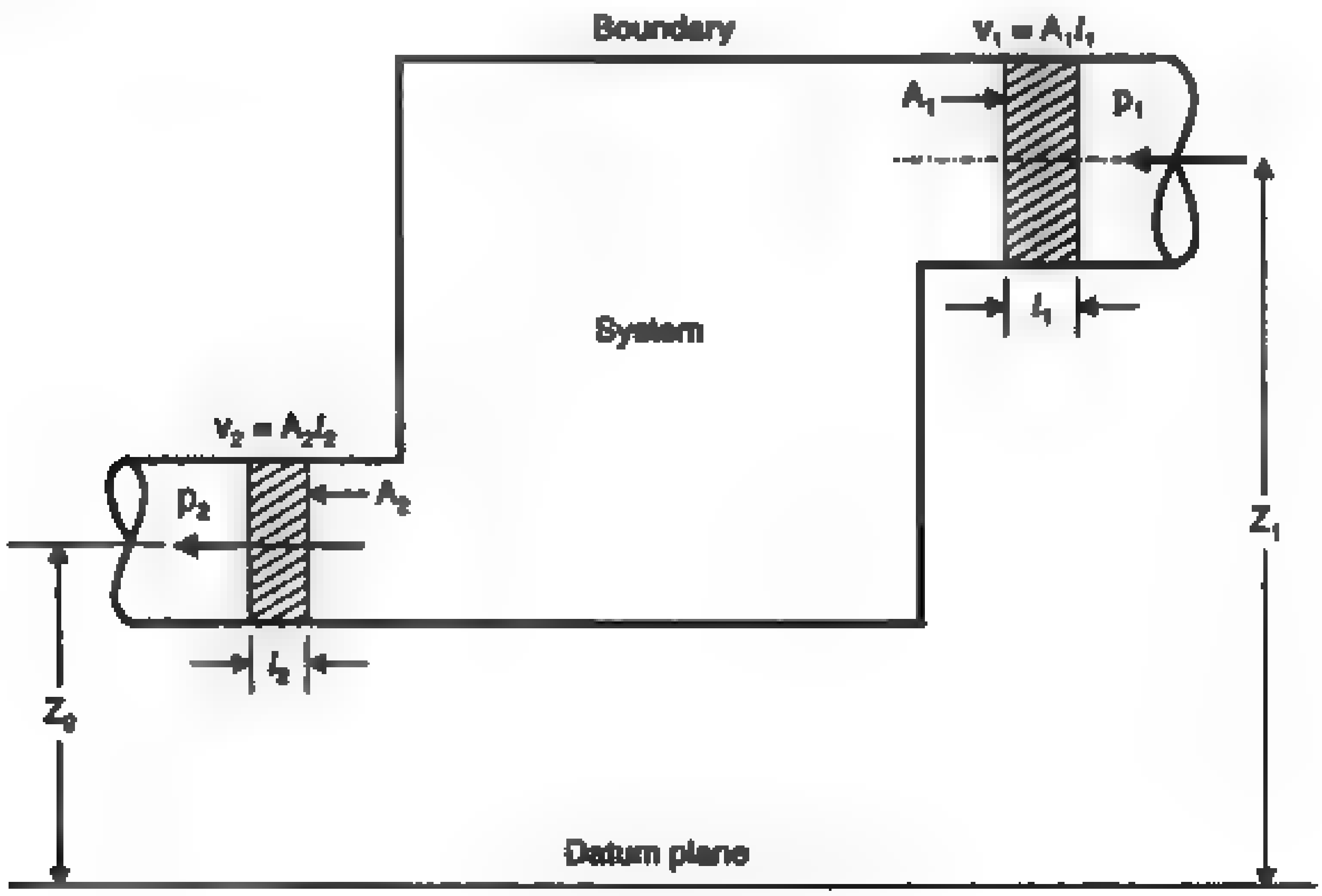


Fig. 1.11

The steady flow equation can be expressed as follows :

$$u_1 + \frac{C_1^2}{2} + Z_1 g + p_1 v_1 + Q = u_2 + \frac{C_2^2}{2} + Z_2 g + p_2 v_2 + W \quad ...(1.16)$$

$$(u_1 + p_1 v_1) + \frac{C_1^2}{2} + Z_1 g + Q = (u_2 + p_2 v_2) + \frac{C_2^2}{2} + Z_2 g + W$$

$$h_1 + \frac{C_1^2}{2} + Z_1 g + Q = h_2 + \frac{C_2^2}{2} + Z_2 g + W \quad [\because \ h = u + pv]$$

If Z_1 and Z_2 are neglected, we get

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W \quad ...[1.16\ (a)]$$

where Q = Heat supplied (or entering the boundary) per kg of fluid ;
W = Work done by (or work coming out of the boundary) 1 kg of fluid ;
C = Velocity of fluid ;
Z = Height above datum ;
p = Pressure of the fluid ;
u = Internal energy per kg of fluid ;
pv = Energy required for 1 kg of fluid.

This equation is applicable to any medium in any steady flow. It is applicable not only to rotary machines such as centrifugal fans, pumps and compressors but also to reciprocating machines such as steam engines.

In a steady flow the rate of mass flow of fluid at any section is the same as at any other section. Consider any section of cross-sectional area A, where the fluid velocity is C, the rate of volume flow past the section is CA. Also, since mass flow is volume flow divided by specific volume,

$$\text{Mass flow rate, } \dot{m} = \frac{CA}{v} \quad ...(1.17)$$

(where v = *specific volume* at the section)

This equation is known as the **continuity of mass equation.**

With reference to Fig. 1.11.

$$\therefore \quad \dot{m} = \frac{C_1 A_1}{v_1} = \frac{C_2 A_2}{v_2} \quad ...[1.17\ (a)]$$

1.16.1. Energy Relations for Flow Process

The energy equation (m kg of fluid) for a steady flow system is given as follows :

$$m\left(u_1 + \frac{C_1^2}{2} + Z_1 g + p_1 v_1\right) + Q = m\left(u_2 + \frac{C_2^2}{2} + Z_2 g + p_2 v_2\right) + W$$

i.e.,
$$Q = m\left[(u_2 - u_1) + (Z_2 g - Z_1 g) + \left(\frac{C_2^2}{2} - \frac{C_1^2}{2}\right) + (p_2 v_2 - p_1 v_1)\right] + W$$

i.e.,
$$Q = m\left[(u_2 - u_1) + g(Z_2 - Z_1) + \left(\frac{C_2^2 - C_1^2}{2}\right) + (p_2 v_2 - p_1 v_1)\right] + W$$

$$= \Delta U + \Delta PE + \Delta KE + \Delta (pv) + W$$

where
$$\Delta U = m\,(u_2 - u_1)$$
$$\Delta PE = mg\,(Z_2 - Z_1)$$
$$\Delta KE = m\left(\frac{C_2^2 - C_1^2}{2}\right)$$
$$\Delta pv = m(p_2v_2 - p_1v_1)$$

$\therefore$
$$Q - \Delta U = [\Delta PE + \Delta KE + \Delta(pv) + W] \qquad ...(1.18)$$

For non-flow process,
$$Q = \Delta U + W = \Delta U + \int_1^2 pdV$$

i.e.,
$$Q - \Delta U = \int_1^2 pdV \qquad ...(1.19)$$

1.19. LIMITATIONS OF FIRST LAW OF THERMODYNAMICS

It has been observed that *energy can flow* from a system in the form of *heat* or *work*. The first law of thermodynamics sets no limit to the amount of the total energy of a system which can be caused to flow out as work. A limit is imposed, however, as a result of the principle enunciated in the second law of thermodynamics which states that heat will flow naturally from one energy reservoir to another at a lower temperature, but not in opposite direction without assistance. This is very important because a heat engine operates between two energy reservoirs at different temperatures

Further the first law of thermodynamics *establishes equivalence between the quantity of heat used and the mechanical work but does not specify the conditions under which conversion of heat into work is possible, neither the direction in which heat transfer can take place.* This gap has been *bridged* by the second law of thermodynamics.

1.20. PERFORMANCE OF HEAT ENGINE AND REVERSED HEAT ENGINE

Refer Fig. 1.12 (*a*). A *heat engine* is used to produce the maximum work transfer from a given positive heat transfer. The measure of success is called the *thermal efficiency* of the engine and is defined by the ratio :

$$\text{Thermal efficiency, } \eta_{th} = \frac{W}{Q_1} \qquad ...(1.20)$$

where W = Net work transfer from the engine, and
Q_1 = Heat transfer to engine.

For a *reversed heat engine* [Fig. 1.12 (*b*)] acting as a *refrigerator* when the purpose is to achieve the maximum heat transfer from the cold reservoir, the measure of success is called the co-efficient of performance (C.O.P.). It is defined by the ratio :

$$\text{Co-efficient of performance, } (\text{C.O.P.})_{ref.} = \frac{Q_2}{W} \qquad ...(1.21)$$

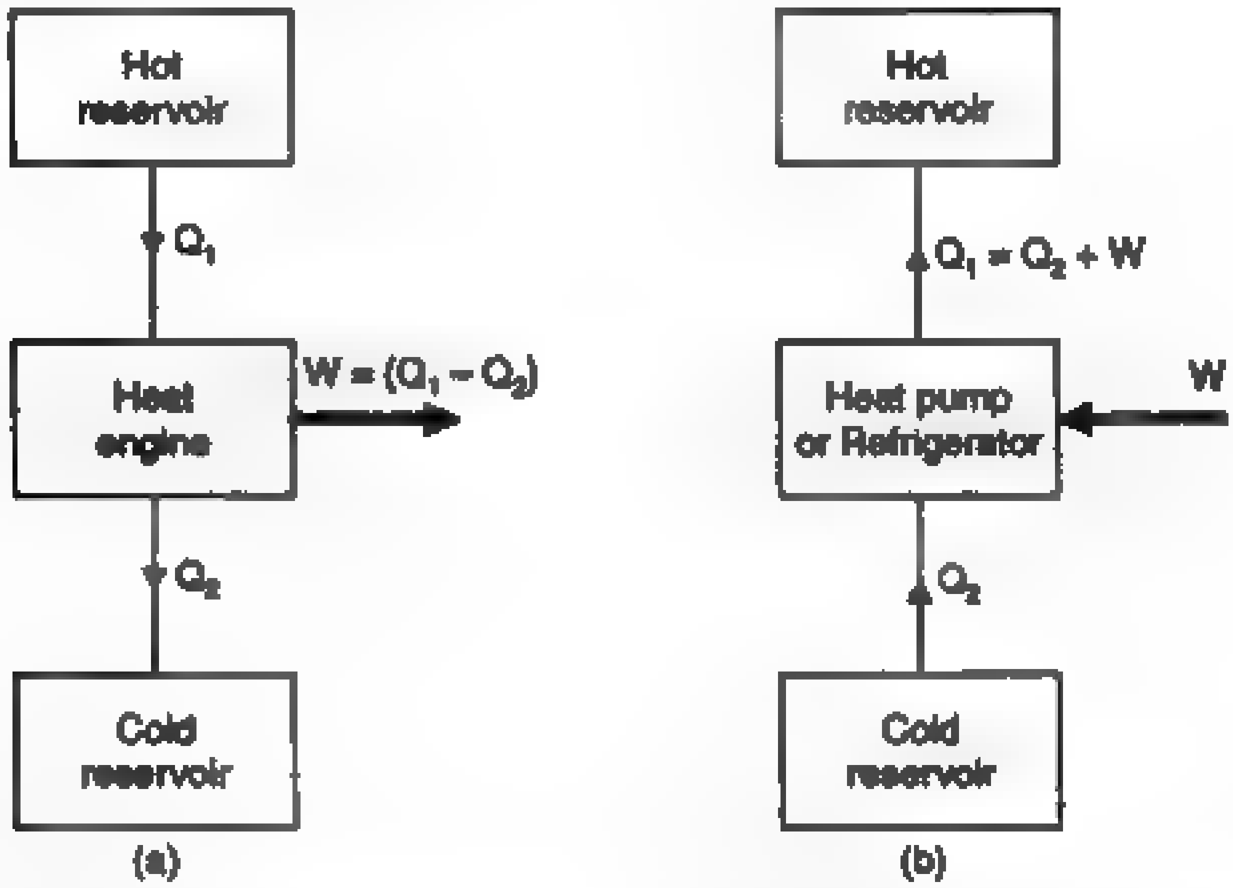
where Q_2 = Heat transfer *from cold reservoir*
W = The net work transfer to the refrigerator.

For a **reversed heat engine** [Fig. 1.12 (*b*)] acting as a *heat pump*, the measure of success is again called the *co-efficient of performance*. It is defined by the ratio :

$$\text{Co-efficient of performance, } (\text{C.O.P.})_{heat\ pump} = \frac{Q_1}{W} \quad ...(1.22)$$

where Q_1 = Heat transfer *to hot reservoir*

W = Net work transfer to the heat pump.



(*a*) Heat engine (*b*) Heat pump or refrigerator.

Fig. 1.12

In all the above three cases application of the first law gives the relation $Q_1 - Q_2 = W$, and this can be used to rewrite the expressions for thermal efficiency and co-efficient of performance solely in terms of the heat transfers.

$$\eta_{th} = \frac{Q_1 - Q_2}{Q_1} \quad ...(1.23)$$

$$(C.O.P.)_{ref} = \frac{Q_2}{Q_1 - Q_2} \quad ...(1.24)$$

$$(C.O.P.)_{heat\ pump} = \frac{Q_1}{Q_1 - Q_2} \quad ...(1.25)$$

It may be seen that η_{th} is *always less than unity* and $(C.O.P.)_{heat\ pump}$ is *always greater than unity*.

1.21. STATEMENTS OF SECOND LAW OF THERMODYNAMICS

The second law of thermodynamics has been enunciated meticulously by Clausius, Kelvin and Planck in slightly different words although both statements are basically identical. Each statement is based on an *irreversible process*. The *first considers transformation of heat between two thermal reservoirs* while the *second considers the transformation of heat into work*.

1.21.1. Clausius Statement

"It is impossible for a self acting machine working in a cyclic process unaided by any external agency, to convey heat from a body at a lower temperature to a body at a higher temperature".

In other words, heat of, itself, cannot flow from a colder to a hotter body.

1.21.2. Kelvin-Planck Statement

"It is impossible to construct an engine, which while operating in a cycle produces no other effect except to extract heat from a single reservoir and do equivalent amount of work".

Although the Clausius and Kelvin-Planck statements appear to be different, they are really equivalent in the sense that a *violation of either statement implies violation of other.*

1.22. ENTROPY

1.22.1. Introduction

In heat engine theory, the term entropy plays a vital role and leads to important results which by other methods can be obtained much more laboriously.

It may be noted that all heat is not equally valuable for converting into work. Heat that is supplied to a substance at high temperature has a greater possibility of conversion into work than heat supplied to a substance at a lower temperature.

*"**Entropy** is a function of a quantity of heat which shows the possibility of conversion of that heat into work. The increase in entropy is small when heat is added at a high temperature and is greater when heat addition is made at a lower temperature. Thus for maximum entropy, there is minimum availability for conversion into work and for minimum entropy there is maximum availability for conversion into work."*

The entropy attains its maximum value when the system reaches a stable equilibrium state from a non-equilibrium state. This is the state of maximum disorder and is one of maximum thermodynamic probability.

1.22.2. Temperature-Entropy Diagram

If entropy is plotted horizontally and absolute temperature vertically, the diagram so obtained is called *temperature entropy* (T-s) diagram. Such a diagram is shown in Fig. 1.13. If working

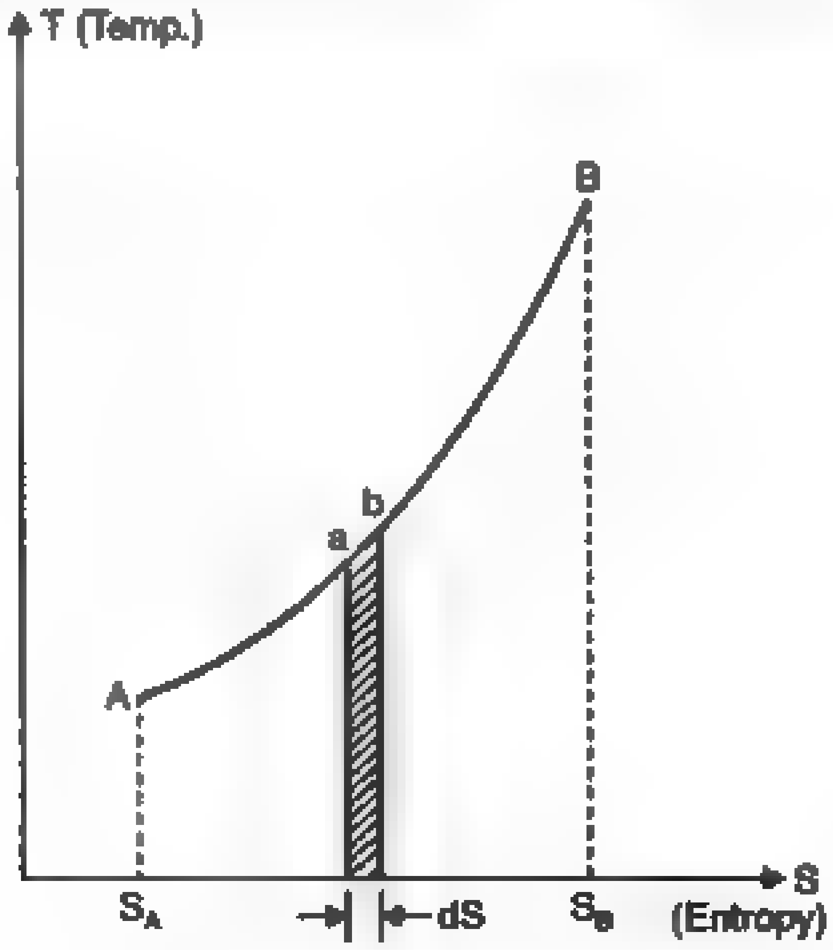


Fig. 1.13. Temperature-entropy diagram.

fluid receives a small amount of heat dQ in an elementary portion ab of an operation AB when temperature is T, and if dQ is represented by the shaded area of which T is the mean ordinate, the width of the figure must be $\frac{dQ}{T}$. This is called *'increment of entropy'* and is denoted by dS. The total heat received by the operation will be given by the area under the curve AB and $(S_B - S_A)$ will be corresponding increase of entropy.

From above we conclude that :

$$\text{Entropy change,} \quad dS = \frac{\text{Heat Change } (Q)}{\text{Absolute temperature } (T)} .$$

***"Entropy** may also be defined as the thermal property of a substance which remains constant when substance is expanded or compressed adiabatically in a cylinder".*

Note. 's' stands for specific entropy whereas 'S' means total entropy (*i.e.*, $S = ms$).

1.22.3. Characteristics of Entropy

The *characteristics of entropy* in a summarised form are given below :

1. It increases when heat is supplied irrespective of the fact whether temperature changes or not.
2. It decreases when heat is removed whether temperature changes or not.
3. It remains unchanged in all adiabatic frictionless processes.
4. It increases if temperature of heat is lowered without work being done as in a throttling process.

Table 1.2. Summary of Formulae

S. No.	*Process*	*Change of entropy (per kg)*
1.	General case	(*i*) $c_v \log_e \frac{T_2}{T_1} + R \log_e \frac{v_2}{v_1}$ (in terms of T and v)
		(*ii*) $c_v \log_e \frac{p_2}{p_1} + c_p \log_e \frac{v_2}{v_1}$ (in terms of p and v)
		(*iii*) $c_p \log_e \frac{T_2}{T_1} - R \log_e \frac{p_2}{p_1}$ (in terms of T and p)
2.	Constant volume	$c_v \log_e \frac{T_2}{T_1}$
3.	Constant pressure	$c_p \log_e \frac{T_2}{T_1}$
4.	Isothermal	$R \log_e \frac{v_2}{v_1}$
5.	Adiabatic	Zero
6.	Polytropic	$c_v \left(\frac{n-\gamma}{n-1}\right) \log_e \frac{T_2}{T_1}$

1.23. THE THIRD LAW OF THERMODYNAMICS

The third law of thermodynamics is stated as follows :

"The entropy of all perfect crystalline solids is zero at absolute zero temperature".

The third law of thermodynamics, often referred to as **Nernst law**, provides the basis for the calculations of absolute entropies of substances.

According to this law, if the entropy is zero at $T = 0$, the absolute entropy $s_{ab.}$ of a substance at any temperature T and pressure p is expressed by the expression

$$s_{ab.} = \int_0^{T_s = T_{f_1}} c_{ps} \frac{dT}{T} + \frac{h_{sf}}{T_s} + \int_{T_1}^{T_{f_2} = T_g} c_{pf} \frac{dT}{T} + \frac{h_{fg}}{T_g} \int_{T_g}^{T} c_{pg} \frac{dT}{T} \quad \text{...(1.26)}$$

where $\quad T_s = T_{f_1} = T_{sf} = T_{sat}$ ...for fusion,

$T_{f_2} = T_g = T_{fg} = T_{sat}$ ...for vaporisation,

c_{ps}, c_{pf}, c_{pg} = Constant pressure specific heats for solids, liquids and gas, and

h_{sf}, h_{fg} = Latent heats of fusion and vaporisation.

Thus by putting $s = 0$ at $T = 0$, one may integrate zero kelvin and standard state of 298.15 K and 1 atm., and find the entropy difference.

Further, it can be shown that the entropy of a crystalline substance at $T = 0$ is not a function of pressure, *viz.*, $\left(\frac{\partial s}{\partial p}\right)_{T=0} = 0$.

However, at temperature above absolute zero, the entropy is a function of pressure also.

1.24. AVAILABLE AND UNAVAILABLE ENERGY

There are many forms in which an energy can exist. But even under ideal conditions all these forms cannot be converted completely into work. This indicates that energy has two parts :

— Available part.

— Unavailable part.

'Available energy' is the *maximum portion of energy which could be converted into useful work by ideal processes which reduce the system to a dead state* (a state in equilibrium with the earth and its atmosphere). Because there can be only one value for maximum work which the system alone could do while descending to its dead state, it follows immediately that *'Available energy' is a property.*

A system which has a pressure difference from that of surroundings, work can be obtained from an expansion process, and if the system has a different temperature, heat can be transferred to a cycle and work can be obtained. But when the temperature and pressure becomes equal to that of the earth, transfer of energy ceases, and although the system contains internal energy, this energy is *unavailable*.

Summarily available energy denote, the latent capability of energy to do work, and in this sense it can be applied to energy in the system or in the surroundings.

The theoretical maximum amount of work which can be obtained from a system at any state p_1 and T_1 when operating with a reservoir at the constant pressure and temperature p_0 and T_0 is called **'availability'.**

HIGHLIGHTS

1. *Thermodynamics* is an axiomatic science which deals with the relations among heat, work and properties of systems which are in equilibrium. It basically entails *four* laws or axioms known as *Zeroth, First, Second* and *Third* law of thermodynamics.
2. A *system* is a finite quantity of matter or a prescribed region of space.
   A system may be a *closed, open* or *isolated* system.
3. A *phase* is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.
4. A *homogeneous system* is one which consists of a *single phase*.
5. A *heterogeneous system* is one which consists of *two or more phases*.
6. A *pure substance* is one that has a homogeneous and invariable chemical composition even though there is a change of phase.
7. A system is in *thermodynamic equilibrium* if temperature and pressure at all points are same ; there should be no *velocity gradient*.
8. A *property of a system* is a characteristic of the system which depends upon its state, but not upon how the state is reached.
   Intensive properties do not depend on the mass of the system.
   Extensive properties depend on the mass of the system.
9. *State* is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a state.
10. A *process* occurs when the system undergoes a change in state or an energy transfer takes place at a steady [illegible]
11. Any process or series of processes whose end states are identical is termed a *cycle*.
12. The *pressure* of a system is the force exerted by the system on unit area of boundaries. Vacuum is defined as the absence of pressure.
13. A *reversible process* is one which can be stopped at any stage and reversed so that the system and surroundings are exactly restored to their initial states.
    An *irreversible process* is one in which heat is transferred through a finite temperature.
14. Zeroth law of termodynamics states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.
15. Infinite slowness is the characteristic feature of a quasi-static process. A quasi-static process is a succession of equilibrium states. It is also called a reversible process.
16. *Internal energy* is the heat energy stored in a gas. The internal energy of a perfect gas is a function of *temperature* only.
17. First law of thermodynamics states :
    — Heat and work are mutually convertible but since energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant.

    Or

    — No machine can produce energy without corresponding expenditure of energy, *i.e.* it is impossible to construct a perpetual motion machine of first kind.

    First law can be expressed as follows :

    $$Q = \Delta E + W$$

    $$Q = \Delta U + W$$ ... if electric, magnetic, chemical energies are absent and changes in potential and kinetic energies are neglected.

18. **There can be no machine which would continuously supply mechanical work without some form of energy disappearing simultaneously. Such a fictitious machine is called a perpetual motion machine of the first kind, or in brief, PMM1. A PMM1 is thus impossible.**
19. The energy of an isolated system is always constant.
20. In case of

(*i*) **Reversible constant volume process** (v = constant)

$$\Delta u = c_v(T_2 - T_1)\ ;\ W = 0\ ;\ Q = c_v\,(T_2 - T_1)$$

(*ii*) **Reversible constant pressure process** (p = constant)

$$\Delta u = c_v(T_2 - T_1)\ ;\ W = p(v_2 - v_1)\ ;\ Q = c_p\,(T_2 - T_1)$$

(*iii*) **Reversible temperature or isothermal process** (pv = constant)

$$\Delta u = 0,\ W = p_1V_1 \log_e r,\ Q = W$$

where r = expansion or compression ratio.

(*iv*) **Reversible adiabatic process** (pv^γ = constant)

$$\pm\,\Delta u = \mp\ W = \frac{R(T_1 - T_2)}{\gamma - 1}\ ;\ Q = 0\ ;\ \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

(*v*) **Polytropic reversible process** (pv^n = constant)

$$\Delta u = c_v\,(T_2 - T_1)\ ;\ W = \frac{R(T_1 - T_2)}{n - 1}\ ;\ Q = \Delta u + W\ ;$$

and $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \quad \text{and} \quad Q = \left(\frac{\gamma - n}{\gamma - 1}\right) \times W.$$

21. Steady flow equation can be expressed as follows :

$$u_1 + \frac{C_1^2}{2} + Z_1 g + p_1v_1 + Q = u_2 + \frac{C_2^2}{2} + Z_2 g + p_2v_2 + W \qquad ...(i)$$

or $$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W, \text{ neglecting } Z_1 \text{ and } Z_2 \qquad ...(ii)$$

where Q = Heat supplied per kg of fluid ; W = Work done by 1 kg of fluid ;
C = Velocity of fluid ; Z = Height above datum ;
p = Pressure of the fluid ; u = Internal energy per kg of fluid ;
pv = Energy required per kg of fluid.

This equation is applicable to any medium in any steady flow.

22. **Clausius statement :**

"It is impossible for a self-acting machine working in a cyclic process, unaided by any external agency, to convey heat from a body at a lower temperature to a body at a higher temperature."

Kelvin-Planck statement :

"It is impossible to construct an engine, which while operating in a cycle produces no other effect except to extract heat from a single reservoir and do equivalent amount of work".

Although above statements of second law of thermodynamics appear to be different, they are really equivalent in the sense that violation of either statement implies violation of other.

23. Clausius inequality is given by,

$$\sum_{\text{Cycle}} \left(\frac{\delta Q}{T}\right) \le 0$$

"When a system performs a reversible cycle, then

$$\sum_{\text{Cycle}} \left(\frac{\delta Q}{T}\right) = 0,$$

but when the cycle is not reversible

$$\sum_{\text{Cycle}} \left(\frac{\delta Q}{T}\right) < 0."$$

24. **'Entropy'** is a function of a quantity of heat which shows the possibility of conversion of that heat into work. The increase in entropy is small when heat is added at a high temperature and is greater when heat addition is made at lower temperature. Thus for maximum entropy, there is a minimum availability for conversion into work and for minimum entropy there is maximum availability for conversion into work.
25. **The third law of thermodynamics** is stated as follows :
"The entropy of all perfect crystalline solids is zero at absolute zero temperature".

OBJECTIVE TYPE QUESTIONS

Choose the correct answer :

1. A definite area or space where some thermodynamic process takes place is known as
(*a*) thermodynamic system (*b*) thermodynamic cycle
(*c*) thermodynamic process (*d*) thermodynamic law.
2. An open system is one in which
(*a*) heat and work cross the boundary of the system, but the mass of the working substance does not
(*b*) mass of working substance crosses the boundary of the system but the heat and work do not
(*c*) both the heat and work as well as mass of the working substances cross the boundary of the system
(*d*) neither the heat and work nor the mass of the working substances cross the boundary of the system.
3. An isolated system
(*a*) is a specified region where transfer of energy and/or mass take place
(*b*) is a region of constant mass and only energy is allowed to cross the boundaries
(*c*) cannot transfer either energy or mass to or from the surroundings
(*d*) is one in which mass within the system is not necessarily constant
(*e*) none of the above.
4. In an extensive property of a thermodynamic system
(*a*) extensive heat is transferred (*b*) extensive work is done
(*c*) extensive energy is utilised (*d*) all of the above
(*e*) none of the above.
5. Which of the following is an intensive property of a thermodynamic system ?
(*a*) Volume (*b*) Temperature
(*c*) Mass (*d*) Energy.
6. Which of the following is the extensive property of a thermodynamic system ?
(*a*) Pressure (*b*) Volume
(*c*) Temperature (*d*) Density.
7. When two bodies are in thermal equilibrium with a third body they are also in thermal equilibrium with each other. This statement is called
(*a*) Zeroth law of thermodynamics (*b*) First law of thermodynamics
(*c*) Second law of thermodynamics (*d*) Kelvin-Planck's law.

8. The temperature at which the volume of a gas becomes zero is called
(a) absolute scale of temperature (b) absolute zero temperature
(c) absolute temperature (d) none of the above.
9. The value of one bar (in SI units) is equal to
(a) 100 N/m^2 (b) 1000 N/m^2 (c) 1×10^4 N/m^2
(d) 1×10^5 N/m^2 (e) 1×10^6 N/m^2.
10. The absolute zero pressure will be
(a) when molecular momentum of the system becomes zero
(b) at sea level (c) at the temperature of – 273 K
(d) under vacuum conditions (e) at the centre of the earth.
11. Absolute zero temperature is taken as
(a) – 273°C (b) 273°C
(c) 237°C (d) – 373°C.
12. Which of the following is correct ?
(a) Absolute pressure = gauge pressure + atmospheric pressure
(b) Gauge pressure = absolute pressure + atmospheric pressure
(c) Atmospheric pressure = absolute pressure + gauge pressure
(d) Absolute pressure = gauge pressure – atmospheric pressure.
13. The unit of energy in SI units is
(a) Joule (J) (b) Joule metre (Jm)
(c) Watt (W) (d) Joule/metre (J/m).
14. One watt is equal to
(a) 1 Nm/s (b) 1 N/min (c) 10 N/s
(d) 100 Nm/s (e) 100 Nm/m.
15. One joule (J) is equal to
(a) 1 Nm (b) kNm
(c) 10 Nm/s (d) 10 kNm/s.
16. The amount of heat required to raise the temperature of 1 kg of water through 1°C is called
(a) specific heat at constant volume (b) specific heat at constant pressure
(c) kilo calorie (d) none of the above.
17. The heating and expanding of a gas is called
(a) thermodynamic system (b) thermodynamic cycle
(c) thermodynamic process (d) thermodynamic law.
18. A series of operations, which take place in a certain order and restore the initial condition is known as
(a) reversible cycle (b) irreversible cycle
(c) thermodynamic cycle (d) none of the above.
19. The condition for the reversibility of a cycle is
(a) the pressure and temperature of the working substance must not differ, appreciably, from those of the surroundings at any stage in the process
(b) all the processes, taking place in the cycle of operation, must be extremely slow
(c) the working parts of the engine must be friction free
(d) there should be no loss of energy during the cycle of operation
(e) all of the above (f) none of the above.
20. In an irreversible process, there is a
(a) loss of heat (b) no loss of heat
(c) gain of heat (d) no gain of heat.

21. The main cause of the irreversibility is
(a) mechanical and fluid friction
(b) unrestricted expansion
(c) heat transfer with a finite temperature difference
(d) all of the above
(e) none of the above.

22. According to kinetic theory of heat
(a) temperature should rise during boiling
(b) temperature should fall during freezing
(c) at low temperature all bodies are in solid state
(d) at absolute zero there is absolutely no vibration of molecules
(e) none of the above.

23. A system comprising a single phase is called a
(a) closed system
(b) open system
(c) isolated system
(d) homogeneous system
(e) heterogeneous system.

24. If all the variables of a stream are independent of time it is said to be in
(a) steady flow
(b) unsteady flow
(c) uniform flow
(d) closed flow
(e) constant flow.

25. A control volume refers to
(a) a fixed region in space
(b) a specified mass
(c) an isolated system
(d) a reversible process only
(e) a closed system.

26. Internal energy of a perfect gas depends on
(a) temperature, specific heats and pressure
(b) temperature, specific heats and enthalpy
(c) temperature, specific heats and entropy
(d) temperature only.

27. In reversible polytropic process
(a) true heat transfer occurs
(b) the entropy remains constant
(c) the enthalpy remains constant
(d) the internal energy remains constant
(e) the temperature remains constant.

28. An isentropic process is always
(a) irreversible and adiabatic
(b) reversible and isothermal
(c) frictionless and irreversible
(d) reversible and adiabatic
(e) none of the above.

29. The net work done per kg of gas in a polytropic process is equal to
(a) $p_1 v_1 \log_e \frac{v_2}{v_1}$
(b) $p_1 (v_1 - v_2)$
(c) $p_2 \left(v_2 - \frac{v_1}{v_2} \right)$
(d) $\frac{p_1 v_1 - p_2 v_2}{n-1}$
(e) $\frac{p_2 v_1 - p_1 v_2}{n-1}$.

30. Steady flow occurs when
(a) conditions do not change with time at any point
(b) conditions are the same at adjacent points at any instant
(c) conditions change steadily with the time
(d) $\left(\frac{\partial v}{\partial t}\right)$ is constant.

31. A reversible process requires that
(a) there be no heat transfer
(b) newton's law of viscosity be satisfied
(c) temperature of system and surroundings be equal
(d) there be no viscous or coloumb friction in the system
(e) heat transfer occurs from surroundings to system only.

32. The first law of thermodynamics for steady flow
   (a) accounts for all energy entering and leaving a control volume
   (b) is an energy balance for a specified mass of fluid
   (c) is an expression of the conservation of linear momentum
   (d) is primarily concerned with heat transfer.
   (e) is restricted in its application to perfect gases.
33. The characteristic equation of gases $pV = mRT$ holds good for
   (a) monoatomic gases (b) diatomic gas (c) real gases
   (d) ideal gases (e) mixture of gases.
34. A gas which obeys kinetic theory perfectly is known as
   (a) monoatomic gas (b) diatomic gas (c) real gas
   (d) pure gas (e) perfect gas.
35. Work done in a free expansion process is
   (a) zero (b) minimum (c) maximum
   (d) positive (e) negative.
36. Which of the following is not a property of the system ?
   (a) Temperature (b) Pressure (c) Specific volume
   (d) Heat (e) None of the above.
37. In the polytropic process equation pv^n = constant, if $n = 0$, the process is termed as
   (a) constant volume (b) constant pressure (c) constant temperature
   (d) adiabatic (e) isothermal.
38. In the polytropic process equation pv^n = constant, if n is infinitely large, the process is termed as
   (a) constant volume (b) constant pressure (c) constant temperature
   (d) adiabatic (e) isothermal.
39. The processes or systems that do not involve heat are called
   (a) isothermal processes (b) equilibrium processes (c) thermal processes
   (d) steady processes (e) adiabatic processes.
40. In a reversible adiabatic process the ratio (T_1/T_2) is equal to
   (a) $\left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$ (b) $\left(\frac{v_1}{v_2}\right)^{\frac{\gamma-1}{\gamma}}$
   (c) $(v_1 v_2)^{\frac{\gamma-1}{2\gamma}}$. (d) $\left(\frac{v_2}{v_1}\right)^{\gamma}$.
41. In isothermal process
   (a) temperature increases gradually (b) volume remains constant
   (c) pressure remains constant (d) enthalpy change is maximum
   (e) change in internal energy is zero.
42. During throttling process
   (a) internal energy does not change (b) pressure does not change
   (c) entropy does not change (d) enthalpy does not change
   (e) volume change is negligible.
43. When a gas is to be stored, the type of compression that would be ideal is
   (a) isothermal (b) adiabatic (c) polytropic
   (d) constant volume (e) none of the above.
44. If a process can be stopped at any stage and reversed so that the system and surroundings are exactly restored to their initial states, it is known as
   (a) adiabatic process (b) isothermal process (c) ideal process
   (d) frictionless process (e) energyless process.

45. The state of a substance whose evaporation from its liquid state is complete, is known as
(a) vapour (b) perfect gas
(c) air (d) steam.

46. In SI units, the value of the universal gas constant is
(a) 0.8314 J/mole/K (b) 8.314 J/mole/K
(c) 83.14 J/mole/K (d) 831.4 J/mole/K
(e) 8314 J/mole/K.

47. When the gas is heated at constant pressure, the heat supplied
(a) increases the internal energy of the gas (b) increases the temperature of the gas
(c) does some external work during expansion (d) both (b) and (c)
(e) none of the above.

48. The gas constant (R) is equal to the
(a) sum of two specific heats (b) difference of two specific heats
(c) product of two specific heats (d) ratio of two specific heats.

49. The heat absorbed or rejected during a polytropic process is
(a) $\left(\frac{\gamma - n}{\gamma - 1}\right) \times$ work done (b) $\left(\frac{\gamma - n}{\gamma - 1}\right)^2 \times$ work done
(c) $\left(\frac{\gamma - n}{\gamma - 1}\right)^{1/2} \times$ work done (d) $\left(\frac{\gamma - n}{\gamma - 1}\right)^3 \times$ work done.

50. Second law of thermodynamics defines
(a) heat (b) work (c) enthalpy
(d) entropy (e) internal energy.

51. For a reversible adiabatic process, the change in entropy is
(a) zero (b) minimum (c) maximum
(d) infinite (e) unity.

52. For any reversible process, the change in entropy of the system and surroundings is
(a) zero (b) unity (c) negative
(d) positive (e) infinite.

53. For any irreversible process the net entropy change is
(a) zero (b) positive (c) negative
(d) infinite (e) unity.

54. The processes of a Carnot cycle are
(a) two adiabatic and two constant volume
(b) one constant volume and one constant pressure and two isentropics
(c) two adiabatics and two isothermals (d) two constant volumes and two isothermals
(e) two isothermals and two isentropics.

55. Isentropic flow is
(a) irreversible adiabatic flow (b) ideal fluid flow (c) perfect gas flow
(d) frictionless reversible flow (e) reversible adiabatic flow.

56. In a Carnot engine, when the working substance gives heat to the sink
(a) the temperature of the sink increases (b) the temperature of the sink remains the same
(c) the temperature of the source decreases
(d) the temperatures of both the sink and the source decrease
(e) changes depend on the operating conditions.

57. If the temperature of the source is increased, the efficiency of the Carnot engine
(a) decreases (b) increases
(c) does not change (d) will be equal to the efficiency of a practical engine
(e) depends on other factors.

58. The efficiency of an ideal Carnot engine depends on
(a) working substance (b) on the temperature of the source only
(c) on the temperature of the sink only
(d) on the temperatures of both the source and the sink
(e) on the construction of engine.

59. The efficiency of a Carnot engine using an ideal gas as the working substance is
(a) $\frac{T_1 - T_2}{T_1}$ (b) $\frac{T_1}{T_1 - T_2}$ (c) $\frac{T_1 T_2}{T_1 - T_2}$
(d) $\frac{T_1 - T_2}{T_1 T_2}$ (e) $\frac{T_2(T_1 - T_2)}{T_1(T_1 + T_2)}$.

60. In a reversible cycle, the entropy of the system
(a) increases (b) decreases
(c) does not change (d) first increases and then decreases
(e) depends on the properties of working substance.

61. A frictionless heat engine can be 100% efficient only if its exhaust temperature is
(a) equal to its input temperature (b) less than its input temperature
(c) 0°C (d) 0°K (e) –100°C.

62. Kelvin-Planck's law deals with
(a) conservation of energy (b) conservation of heat (c) conservation of mass
(d) conversion of heat into work (e) conversion of work into heat.

63. Which of the following statements is correct according to Clausius statement of second law of thermodynamics ?
(a) It is impossible to transfer heat from a body at a lower temperature to a body at a higher temperature
(b) It is impossible to transfer heat from a body at a lower temperature to a body at a higher temperature, without the aid of an external source.
(c) It is possible to transfer heat from a body at a lower temperature to a body at a higher temperature by using refrigeration cycle
(d) None of the above.

64. According to Kelvin-Planck's statement of second law of thermodynamics
(a) It is impossible to construct an engine working on a cyclic process, whose sole purpose is to convert heat energy into work
(b) It is possible to construct an engine working on a cyclic process, whose sole purpose is to convert the heat energy into work
(c) It is impossible to construct a device which while working in a cyclic process produces no effect other than the transfer of heat from a colder body to a hotter body
(d) When two dissimilar metals are heated at one end and cooled at the other, the e.m.f. developed is proportional to the difference of their temperatures at the two end.
(e) None of the above.

65. The property of a working substance which increases or decreases as the heat is supplied or removed in a reversible manner is known as
(a) enthalpy (b) internal energy
(c) entropy (d) external energy.

66. The entropy may be expressed as a function of
    (a) pressure and temperature (b) temperature and volume
    (c) heat and work (d) all of the above
    (e) none of the above.
67. The change of entropy, when heat is absorbed by the gas is
    (a) positive (b) negative (c) positive or negative.
68. Which of the following statements is *correct* ?
    (a) The increase in entropy is obtained from a given quantity of heat at a low temperature
    (b) The change in entropy may be regarded as a measure of the rate of the availability of heat for transformation into work
    (c) The entropy represents the maximum amount of work obtainable per degree drop in temperature
    (d) All of the above.
69. The condition for the reversibility of a cycle is
    (a) the pressure and temperature of working substance must not differ, appreciably from those of the surroundings at any stage in the process
    (b) all the processes taking place in the cycle of operation, must be extremely slow
    (c) the working parts of the engine must be friction free
    (d) there should be no loss of energy during the cycle of operation
    (e) all of the above.
70. In an irreversible process there is a
    (a) loss of heat (b) no loss of work
    (c) gain of heat (d) no gain of heat.
71. The main cause for the irreversibility is
    (a) mechanical and fluid friction (b) unrestricted expansion
    (c) heat transfer with a finite temperature difference
    (d) all of the above.
72. The efficiency of the Carnot cycle may be increased by
    (a) increasing the highest temperature (b) decreasing the highest temperature
    (c) increasing the lowest temperature (d) decreasing the lowest temperature
    (e) keeping the lowest temperature constant.
73. Which of the following is the *correct* statement ?
    (a) All the reversible engines have the same efficiency
    (b) All the reversible and irreversible engines have the same efficiency
    (c) Irreversible engines have maximum efficiency
    (d) All engines are designed as reversible in order to obtain maximum efficiency.

ANSWERS

1. (a)	2. (c)	3. (c)	4. (e)	5. (b)	6. (b)	7. (a)
8. (b)	9. (d)	10. (a)	11. (a)	12. (a)	13. (a)	14. (a)
15. (a)	16. (c)	17. (b)	18. (c)	19. (e)	20. (a)	21. (d)
22. (d)	23. (d)	24. (a)	25. (a)	26. (d)	27. (a)	28. (d)
29. (d)	30. (a)	31. (d)	32. (a)	33. (c)	34. (e)	35. (a)
36. (d)	37. (b)	38. (a)	39. (e)	40. (a)	41. (e)	42. (d)
43. (a)	44. (c)	45. (b)	46. (e)	47. (d)	48. (b)	49. (a)
50. (d)	51. (a)	52. (a)	53. (b)	54. (e)	55. (e)	56. (b)
57. (b)	58. (d)	59. (a)	60. (c)	61. (d)	62. (d)	63. (b)
64. (e)	65. (e)	66. (a)	67. (a)	68. (d)	69. (e)	70. (a)
71. (d)	72. (d)	73. (a).				

THEORETICAL QUESTIONS

1. Define a thermodynamic system. Differentiate between open system, closed system and an isolated system.
2. How does a homogeneous system differ from a heterogeneous system ?
3. What do you mean by a pure substance ?
4. Explain the following terms :
   (*i*) State, (*ii*) Process, and (*iii*) Cycle.
5. Explain briefly zeroth law of thermodynamics.
6. What is a quasi-static process ?
7. What do you mean by 'reversible work' ?
8. Define 'internal energy' and prove that it is a property of a system.
9. Explain the First Law of Thermodynamics as referred to closed systems undergoing a cyclic change.
10. State the First Law of Thermodynamics and prove that for a non-flow process, it leads to the energy equation $Q = \Delta U + W$
11. What is the mechanical equivalent of heat ? Write down its value when heat is expressed in kJ and work is expressed in N-m.
12. What do you mean by "Perpetual motion machine of first kind-PMM 1" ?
13. Why only in constant pressure non-flow process, the enthalpy change is equal to heat transfer ?
14. Prove that the rate of change of heat interchange per unit change of volume when gas is compressed or expanded is given by $\frac{\gamma - n}{\gamma - 1} \times \frac{pdv}{J}$.
15. Write down the general energy equation for steady flow system and simplify when applied for the following systems :
    (*i*) Centrifugal water pump (*ii*) Reciprocating air compressor
    (*iii*) Steam nozzle (*iv*) Steam turbine
    (*v*) Gas turbine.
16. Explain clearly the difference between a non-flow and a steady flow process.
17. State the limitations of first law of thermodynamics.
18. What is the difference between a heat engine and a reversed heat engine ?
19. Enumerate the conditions which must be fulfilled by a reversible process. Give some examples of ideal reversible processes.
20. What is an irreversible process ? Give some examples of irreversible processes.
21. Give the following statements of second law of thermodynamics.
    (*i*) Clausius statement
    (*ii*) Kelvin-Planck statement.
22. Define heat engine, refrigerator and heat pump.
23. What is the perpetual motion machine of the second kind ?
24. What do you mean by 'Thermodynamic temperature' ?
25. What do you mean by 'Clausius inequality' ?
26. Describe the working of a Carnot cycle.
27. Derive an expression for the efficiency of the reversible heat engine.
28. What do you mean by the term 'Entropy' ?

2

Introduction to Internal Combustion Engines

2.1. Heat engines. 2.2. Development of I.C. engines. 2.3. Classification of I.C. engines. 2.4. Applications of I.C. engines. 2.5. Engine cycle-Energy balance. 2.6. Basic idea of I.C. engines. 2.7. Different parts of I.C. engines. 2.8. Terms connected with I.C. engines. 2.9. Working cycles. 2.10. Indicator diagram. 2.11. Four-stroke cycle engines. 2.12. Two stroke cycle engines. 2.13. Intake for compression ignition engines. 2.14. Comparison of four stroke and two stroke cycle engines. 2.15. Comparison of spark ignition (S.I.) and compression ignition (C.I.) engines. 2.16. Comparison between a petrol engine and a diesel engine. 2.17. How to tell a two stroke cycle engine from a four stroke cycle engine ? Highlights—Objective Type Questions—Theoretical Questions.

2.1. HEAT ENGINES

Any type of engine or machine which derives heat energy from the combustion of fuel or any other source and converts this energy into mechanical work is termed as a ***heat engine.***

Heat engines may be *classified* into two main classes as follows :

1. External Combustion Engines.
2. Internal Combustion Engines.

1. External combustion engines (E.C. engines)

In this case, *combustion of fuel takes place outside the cylinder* as in case of *steam engines* where the heat of combustion is employed to generate steam which is used to move a piston in a cylinder. Other examples of external combustion engines are *hot air engines, steam turbine* and *closed cycle gas turbine*. These engines are generally used for driving locomotives, ships, generation of electric power etc.

2. Internal combustion engines (I.C. engines)

In this case, *combustion of the fuel with oxygen of the air occurs within the cylinder* of the engine. The internal combustion engines group includes engines employing mixtures of combustible gases and air, known as *gas engines*, those *using lighter liquid fuel* or spirit known as *petrol engines* and those using heavier liquid fuels, known as *oil compression ignition or diesel engines.*

The detailed classification of heat engines is given in Fig. 2.1.

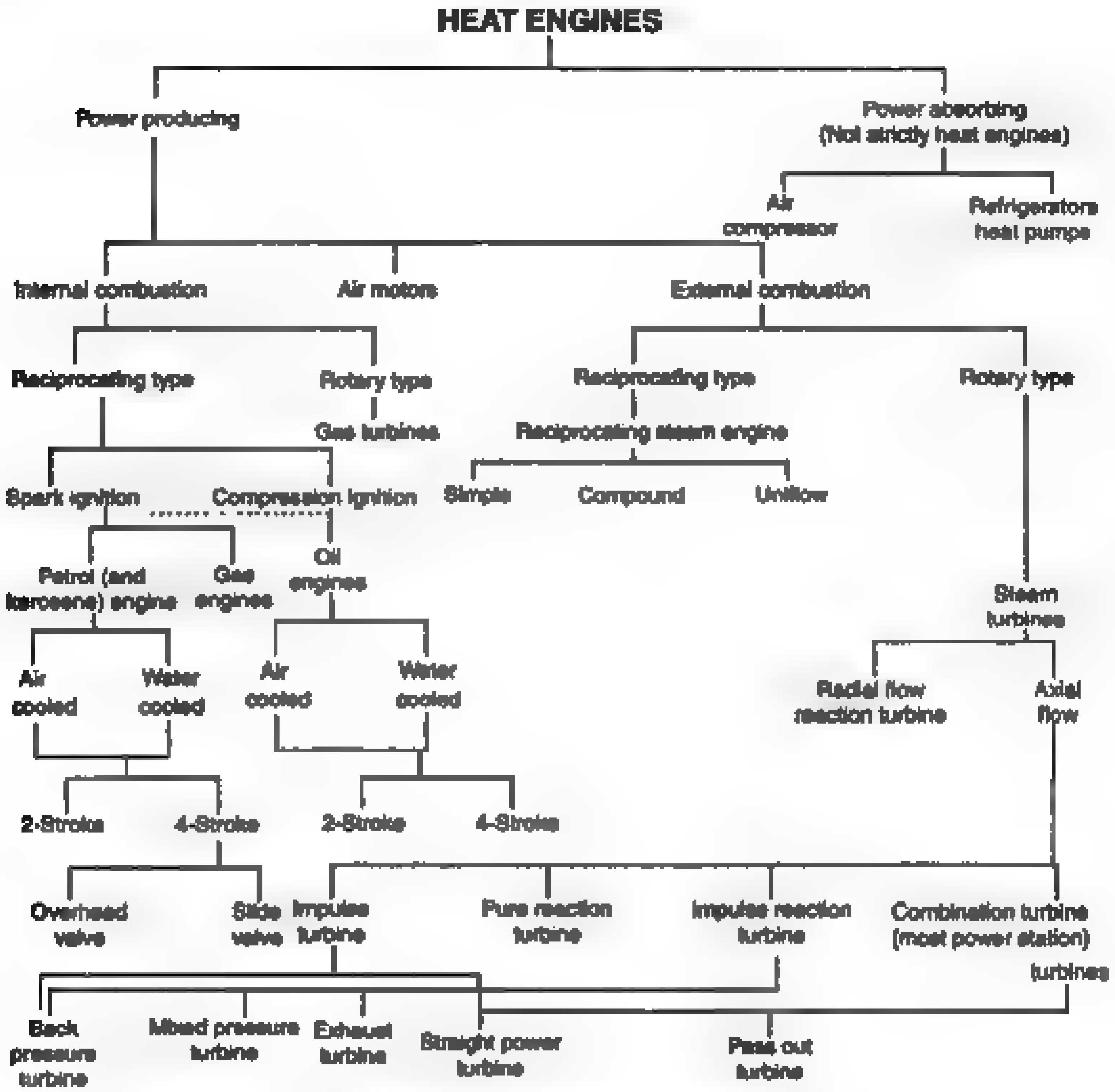


Fig. 2.1. Classification of heat engines.

Advantages of reciprocating internal combustion engines over external combustion engines :

Reciprocating internal combustion engines offer the following *advantages* over external combustion engines :

1. Overall efficiency is high.
2. Greater mechanical simplicity.
3. Weight to power ratio is generally low.
4. Generally lower initial cost.
5. Easy starting from cold conditions.
6. These units are compact and thus require less space.

Advantages of the external combustion engines over internal combustion engines :

The external combustion engines claim the following *advantages* over internal combustion engines :

1. Starting torque is generally high.
2. Because of external combustion of fuel, cheaper fuels can be used. Even solid fuels can be used advantageously.
3. Due to external combustion of fuel it is possible to have flexibility in arrangement.
4. These units are self-starting with the working fluid whereas in case of internal combustion engines, some additional equipment or device is used for starting the engines.

2.2. DEVELOPMENT OF I.C. ENGINES

Brief early history of development of I.C. engines is as follows :

- Many different styles of internal combustion engines were built and tested during the seond half of the 19th century.
- The first fairly practical engine was invented by J.J.E. Lenoir which appeared on the scene about 1860. During the next decade, several hundred of these engines were built with power upto about 4.5 kW and mechanical efficiency upto 5%.
- The Otto-Langen engine with efficiency improved to about 11% was first introduced in 1867 and several thousands of these were produced during the next decade. This was a type of atmospheric engine with the power stroke propelled by atmospheric pressure acting against a vacuum.
- Although many people were working on four-stroke cycle design, Otto was given credit when his prototype engine was built in 1876.
- In the 1880s, the internal combustion engines first appeared in automobiles. Also in this decade the two-stroke cycle engine became practical and was manufactured in large number.
- Rudolf Diesel, by 1892, had perfected his compression ignition engine into basically the same diesel engine known today. This was after years of development work which included the use of solid fuel in his early experimental engines.
- *Early compression engines were noisy, large, slow, single cylinder engines. They were, however, generally more efficient than spark ignition engines.*
- It wasn't until the 1920s that multicylinder compression ignition engines were made small enough to be used with automobile and trucks.
- *Wakdle's first rotary engine* was tested at NSV, Germany in 1957.
- The practical *stirling engines* in small number are being produced since 1965.
  - — These engines require costly material and advanced technology for manufacture.
  - — Thermal efficiencies higher than 30% have been obtained.
  - — The *advantages* of stirling engine *are low exhaust emission and multi-fuel capability.*

2.3. CLASSIFICATION OF I.C. ENGINES

Internal combustion engines may be classified as given below :

1. According to cycle of operation :

(*i*) Two stroke cycle engines

(*ii*) Four stroke cycle engines.

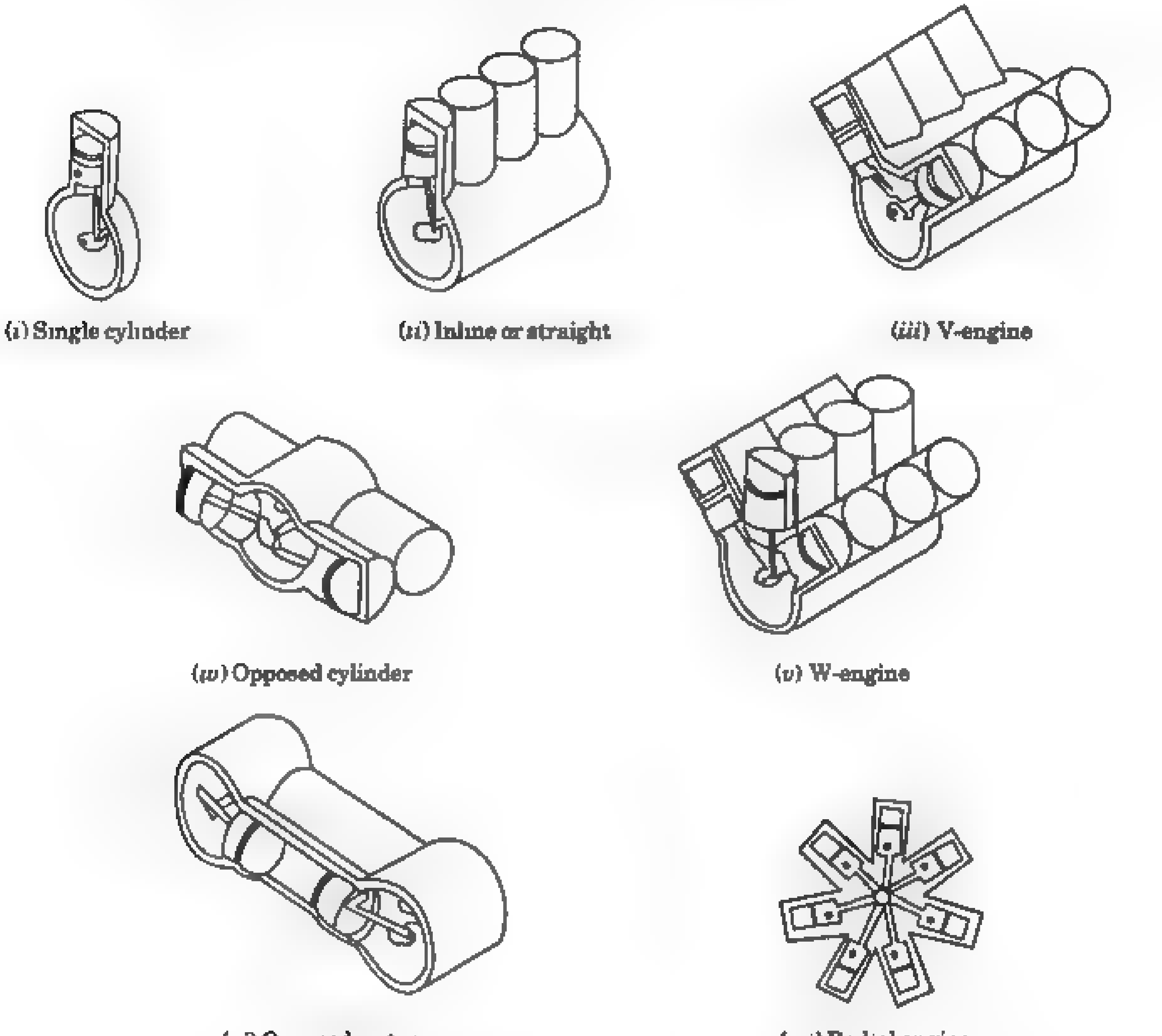
2. According to cycle of combustion :

(*i*) Otto cycle engine (combustion at constant volume)

(*ii*) Diesel cycle engine (combustion at constant pressure)

(*iii*) Dual-combustion or Semi-Diesel cycle engine (combustion partly at constant volume and partly at constant pressure).

3. **According to arrangement of cylinder :** Refer Fig. 2.2.



(*i*) Single cylinder (*ii*) Inline or straight (*iii*) V-engine

(*iv*) Opposed cylinder (*v*) W-engine

(*vi*) Opposed piston (*vii*) Radial engine

Fig. 2.2. Engine classification by cylinder arrangement.

(*i*) **Single cylinder engine.** Engine has one cylinder and piston connected to the crankshaft.

(*ii*) **In-line or straight engines.** Cylinders are positioned in a straight line one behind the other along the length of the crankshaft.

(*iii*) **V-engine**

- An engine with two cylinder banks (*i.e.*, two-in-line engines) inclined at an angle to each other and with one crankshaft.
- Most of the bigger automobiles use the 8-cylinder V-engine (4-cylinder in-line on each side of V).

(*iv*) **Opposed cylinder engine**

- Two banks of cylinders opposite each other on a single crankshaft (a V-engine with 180° V).
- These are common on small aircraft and some automobiles with even number of cylinders from two to eight or more.

(*v*) **W-engine**

- Same as V-engine except with three banks of cylinders on the same crankshaft.
- Not common, but some have been developed for racing automobiles.

(*vi*) **Opposed piston engine**

- In this type of engine there are two pistons in each cylinder with the combustion chamber in the centre between the pistons.
- A single combustion process causes two power strokes, at the same time, with each piston being pushed away from the centre and delivering power to a separate crankshaft at each end of this cylinder.

(*vii*) **Radial engine**

- It is an engine with pistons positioned in a circular plane around the central crankshaft. The connecting rods of the pistons are connected to a master rod which, in turn, is connected to the crankshaft.
- In a radial engine the bank of cylinders always has an *odd* number of cylinders ranging from 3 to 13 or more.
- Operating on a four-stroke cycle, every other cylinder fires and has a power stroke as the crankshaft rotates, giving a smooth operation.
- Many medium and large size propeller-driven aircraft use radial engines. For large aircraft two or more banks of cylinders are mounted together, one behind the other on a single crankshaft, making one powerful smooth engine.
- Very large ship engines exist with upto 54 cylinders, six banks of 9 cylinder each.

4. According to their uses :

(*i*) Stationary engine (*ii*) Portable engine
(*iii*) Marine engine (*iv*) Automobile engine
(*v*) Aero engine etc.

5. According to the speed of the engine :

(*i*) Low speed engine (*ii*) Medium speed engine
(*iii*) High speed engine.

6. According to method of ignition :

(*i*) Spark-ignition engine (*ii*) Compression-ignition engine.

7. According to method of cooling the cylinder :

(*i*) Air-cooled engine (*ii*) Water-cooled engine.

8. According to method of governing :

(*i*) Hit and miss governed engine (*ii*) Quality governed engine
(*iii*) Quantity governed engine.

9. According to valve arrangement :

(*i*) Over head valve engine (*ii*) *L*-head type engine
(*iii*) *T*-head type engine (*iv*) *F*-head type engine.

10. **According to number of cylinders :**

(*i*) Single-cylinder engine (*ii*) Multi-cylinder engine.

11. **According to air intake process :**

(*i*) *Naturally aspirated.* No intake air pressure boost system.

(*ii*) *Supercharged.* Intake air pressure increased with the compressor driven off the engine crankshaft.

(*iii*) *Turbocharged.* Intake air pressure increased with the turbine-compressor driven by the engine exhaust gases.

(*iv*) *Crankcase compressed.* Two stroke-cycle engine which uses the crankcase as the intake air compressor. Limited development work has also been done on the design and construction of four-stroke cycle engines with crankcase compression.

12. **According to fuel employed :**

(*i*) Oil engine (*ii*) Petrol engine
(*iii*) Gas engine (*iv*) Kerosene engine
(*v*) LPG engine (*vi*) Alcohol-ethyl, methyl engine
(*vii*) Duel fuel engine (*viii*) Gasohol (90% gasoline and 10% alcohol).

13. **Method of fuel input for S.I. engines :**

(*i*) *Carburetted*

(*ii*) *Multipoint port fuel injection.* One or more injectors at each cylinder intake.

(*iii*) *Throttle body fuel injection.* Injectors upstream in intake manifold.

2.4. APPLICATION OF I.C. ENGINES

The I.C. engines are generally used for :

(*i*) Road vehicles (*e.g.*, scooter, motorcycle, buses etc.)
(*ii*) Aircraft
(*iii*) Locomotives
(*iv*) Construction in civil engineering equipment such as bull-dozer, scraper, power shovels etc.
(*v*) Pumping sets
(*vi*) Cinemas
(*vii*) Hospital
(*viii*) Several industrial applications.

The *applications of various engines separately* are listed below :

1. **Small two-stroke petrol engines :**

- These engines are employed where *simplicity and the low cost of the prime mover are primary considerations.*
- The 50 c.c. engines develops maximum brake power (B.P.) of 1.5 kW at 5000 r.p.m. and is *used in mopeds.*
- The 100 c.c. engine developing maximum brake power of about 3 kW at 5000 r.p.m. is *used in scooters.* The 150 c.c. engine develops maximum brake power of about 5 kW at 5000 r.p.m.
- The 250 c.c. engine developing a maximum brake power of about 9 kW at 4500 r.p.m. is generally *used in motor cycles.*

- These engines also find applications in very small electric generating sets, pumping sets etc.

2. Small four-stroke petrol engines :

- These engines are primarily used in automobiles.
- These are also used in pumping sets and mobile electric generating sets.

These days diesel engines are taking them over, in the above mentioned applications.

3. Four stroke diesel engines :

- The four-stroke diesel engine (a versatile prime mover) is manufactured in diameter ranging from 50 mm to 600 mm with speeds ranging from 100 to 4400 r.p.m., the power delivered per cylinder varying from 1 to 1000 kW.
- Diesel engine is employed for the following :
  — Pumping sets
  — Construction machinery
  — Air compressors and drilling jigs
  — Tractors
  — Jeeps, cars and taxies
  — Mobile and stationary electric generating plant
  — Diesel-electric locomotive
  — Boats and ships.

4. Two stroke diesel engines :

- These engines having very high power are usually employed for *ship propulsion* and generally have bores above 60 cm, uniflow with exhaust valves or loop scavenged.

Example. Nordberg, *2 stroke, 12-cylinder 80 cm bore and 155 cm stroke, diesel engine engine develops 20000 kW at 120 r.p.m.*

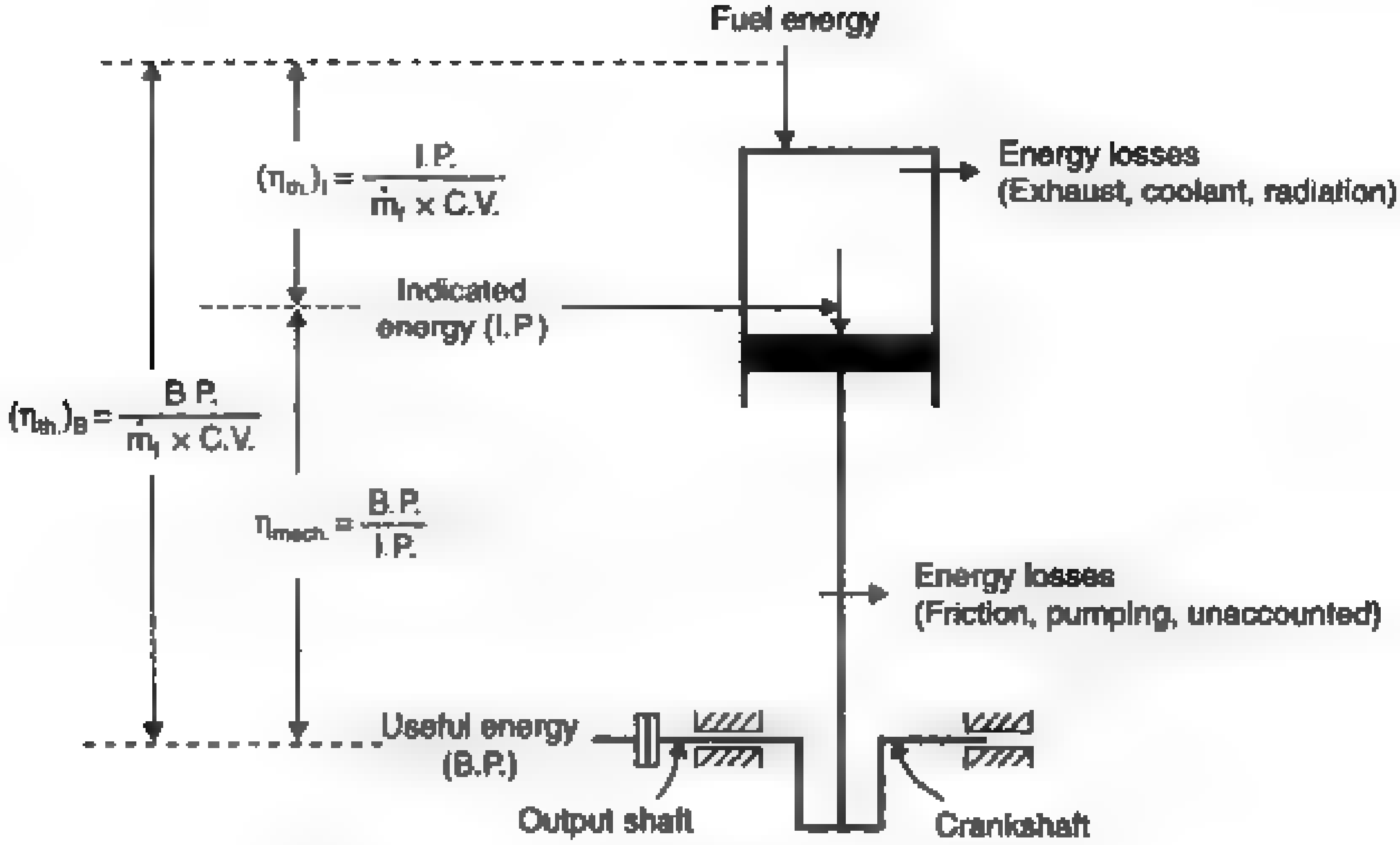
5. Radial piston engine in small aircraft propulsion :

- Radial four stroke petrol engines having power range from 300 kW to 4000 kW have been used in small aircrafts.
- In modern large aircrafts, instead of these engines, gas turbine plant as turboprop engine or turbojet engine and gas turbine engines are used.

2.6. ENGINE CYCLE-ENERGY BALANCE

Refer Fig. 2.3. It shows the energy flow through the reciprocating engine. The analysis is based on the first law of thermodynamics which states that energy can neither be created nor destroyed, it can be converted from one form to other.

- In an I.C. engine fuel is fed to the combustion chamber where it burns in the presence of air and its chemical energy is converted into heat. All this energy is not available for driving the piston since a portion of this energy is lost through exhaust, coolant and radiation. The remaining energy is converted to power and is called indicated energy or *indicated power* (I.P.). The ratio of this energy to the input fuel energy is called indicated thermal efficiency ($\eta_{th.(I)}$).



I.P. = Indicated power

B.P. = Brake power

$(\eta_{th.})_I$ = Indicated thermal efficiency $= \dfrac{I.P.}{m_f \times C.V.}$

(where m_f = mass of fuel in kg/s and C.V. = calorific value)

$(\eta_{th.})_B$ = Brake thermal efficiency.

Fig. 2.3. The energy flow through the reciprocating engine.

- The energy available at the piston passes through the connecting rod to the crankshaft. In this transmission of energy/power there are losses due to friction, pumping, etc. The sum of all these losses, converted to power, is termed as **friction power (F.P.)**. The remaining energy is the *useful mechanical energy and is termed as* **shaft energy** or **brake power (B.P.)**. The *ratio of energy at shaft to fuel input energy is called* **brake thermal efficiency** $[\eta_{th.(B)}]$.
- The *ratio of shaft energy to the energy available at the piston is called* **mechanical efficiency** $(\eta_{mech.})$.

2.6. BASIC IDEA OF I.C. ENGINES

The basic idea of internal combustion engine is shown in Fig. 2.4. The cylinder which is closed at one end is filled with a mixture of fuel and air. As the crankshaft turns it pushes cylinder. The piston is forced up and compresses the mixture in the top of the cylinder. The mixture is set alight and, as it burns, it creates a gas pressure on the piston, forcing it down the cylinder. This motion is shown by arrow '1'. The piston pushes on the rod which pushes on the crank. The crank is given rotary (turning) motion as shown by the arrow '2'. The fly wheel fitted on the end of the crankshaft stores energy and keeps the crank turning steadily.

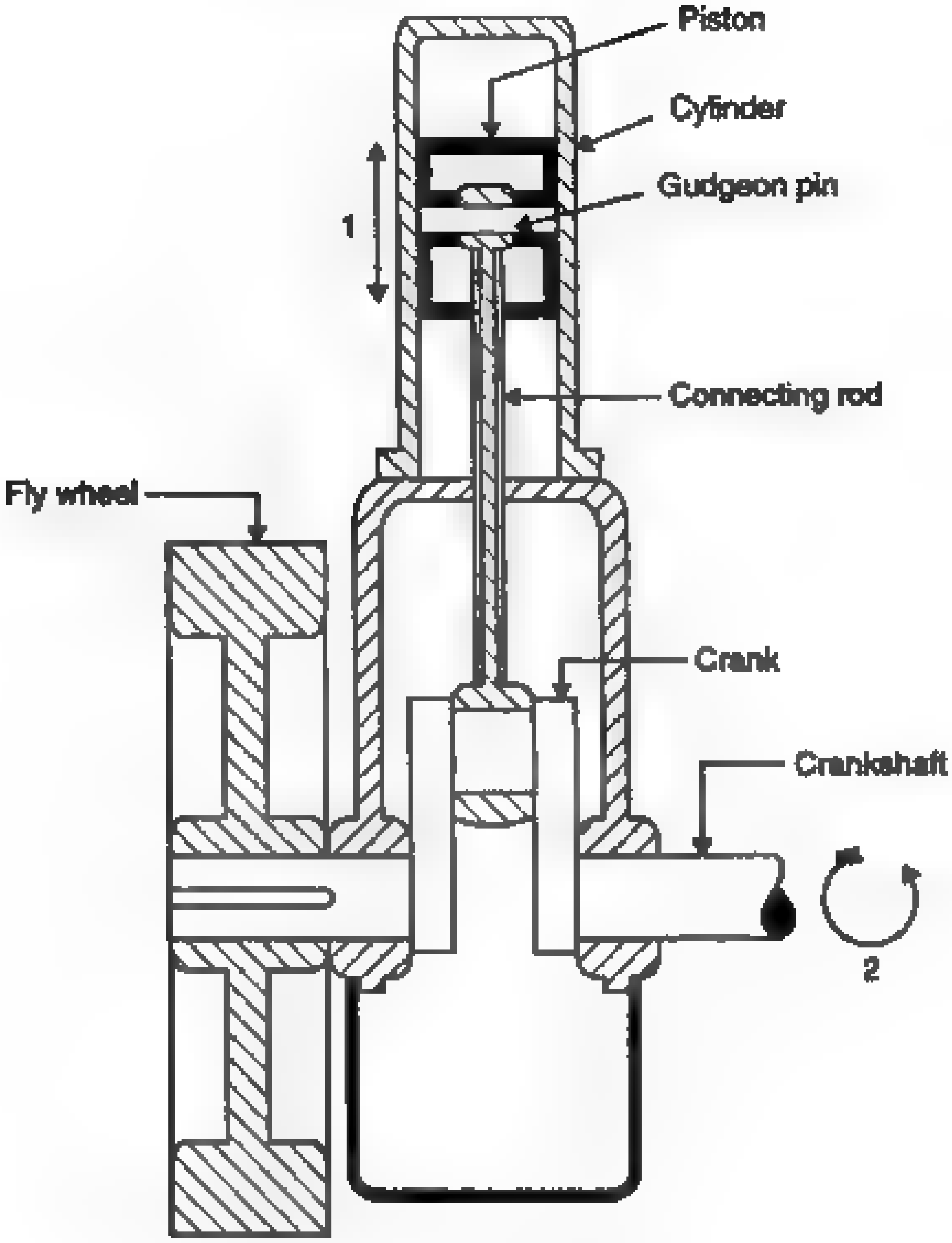


Fig. 2.4. Basic idea of I.C. engine.

2.7. DIFFERENT PARTS OF I.C. ENGINES

Here follows the detail of the various parts of an internal combustion engine.

A cross-section of an air-cooled I.C. engine with principal parts is shown in Fig. 2.5.

A. Parts common to both petrol and diesel engine :

1. Cylinder
2. Cylinder head
3. Piston
4. Piston rings
5. Gudgeon pin
6. Connecting rod
7. Crankshaft
8. Crank
9. Engine bearing
10. Crankcase
11. Flywheel
12. Governor
13. Valves and valve operating mechanisms.

B. Parts for petrol engines only :

1. Spark plugs
2. Carburettor
3. Fuel pump.

C. Parts for Diesel engine only :

1. Fuel pump. 2. Injector.

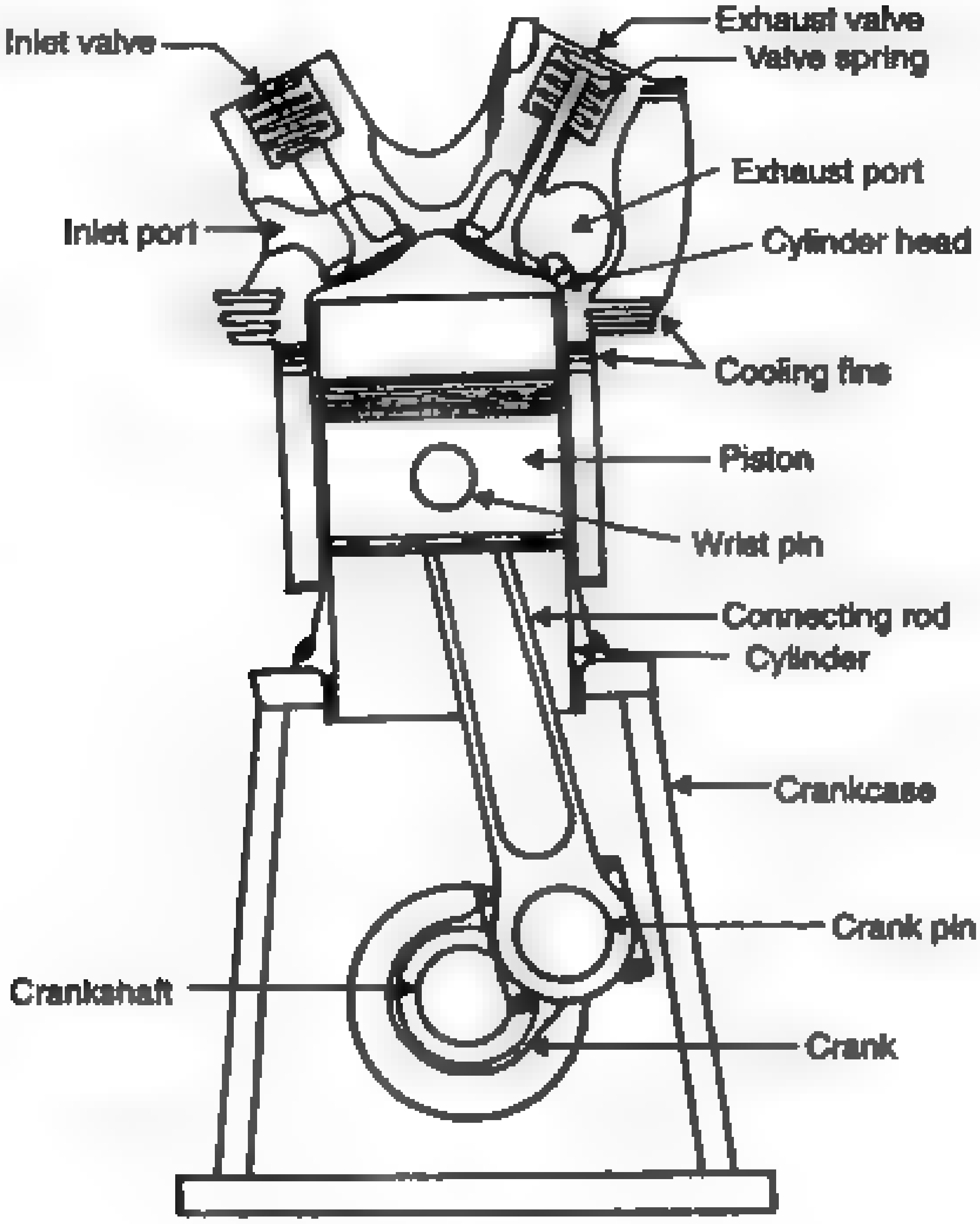


Fig. 2.6. Air-cooled I.C. engine.

A. Parts common to both petrol and diesel engines :

1. Cylinder

The cylinder contains gas under pressure and guides the piston. It is in direct contact with the products of combustion and it must be cooled. The ideal form consists of a plain cylindrical barrel in which the piston slides. The movement of the piston or stroke being in most cases, longer than the bore. This is known as the "*stroke bore ratio*". The upper end consists of a combustion or clearance space in which the ignition and combustion of the charge takes place. In practice, it is necessary to depart from the ideal hemispherical slope in order to accommodate the valves, sparking plugs etc. and to control the combustion. Sections of an air-cooled cylinder and a water-cooled cylinder are shown in Fig. 2.6 and 2.7 respectively. *The cylinder is made of hard grade cast iron and is usually, cast in one piece.*

2 Cylinder head

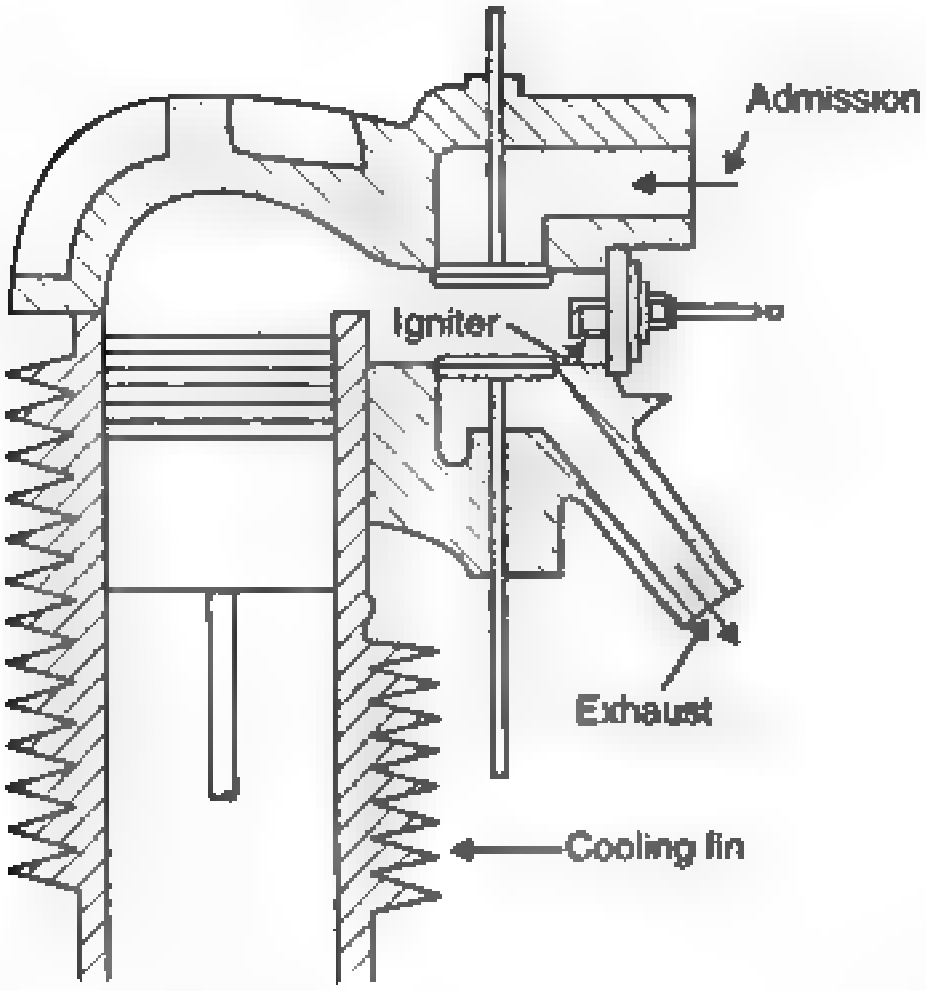


Fig 2.6. Air-cooled cylinder

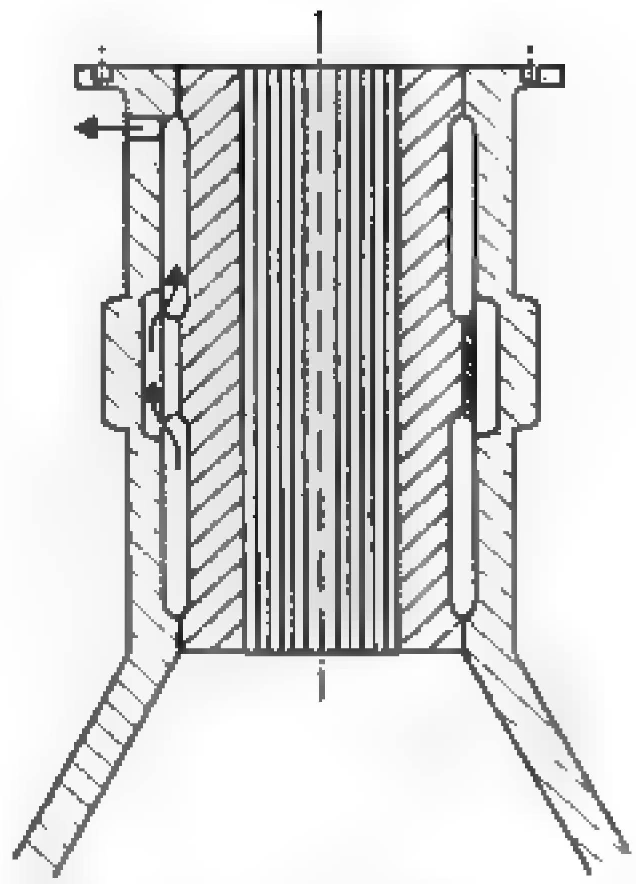


Fig. 2.7 Water-cooled cylinder

One end of the cylinder is closed by means of a *removable cylinder head* (Fig. 2.6) which usually contains the inlet or admission valve [Fig. 2.8 (*a*)] for admitting the mixture of air and

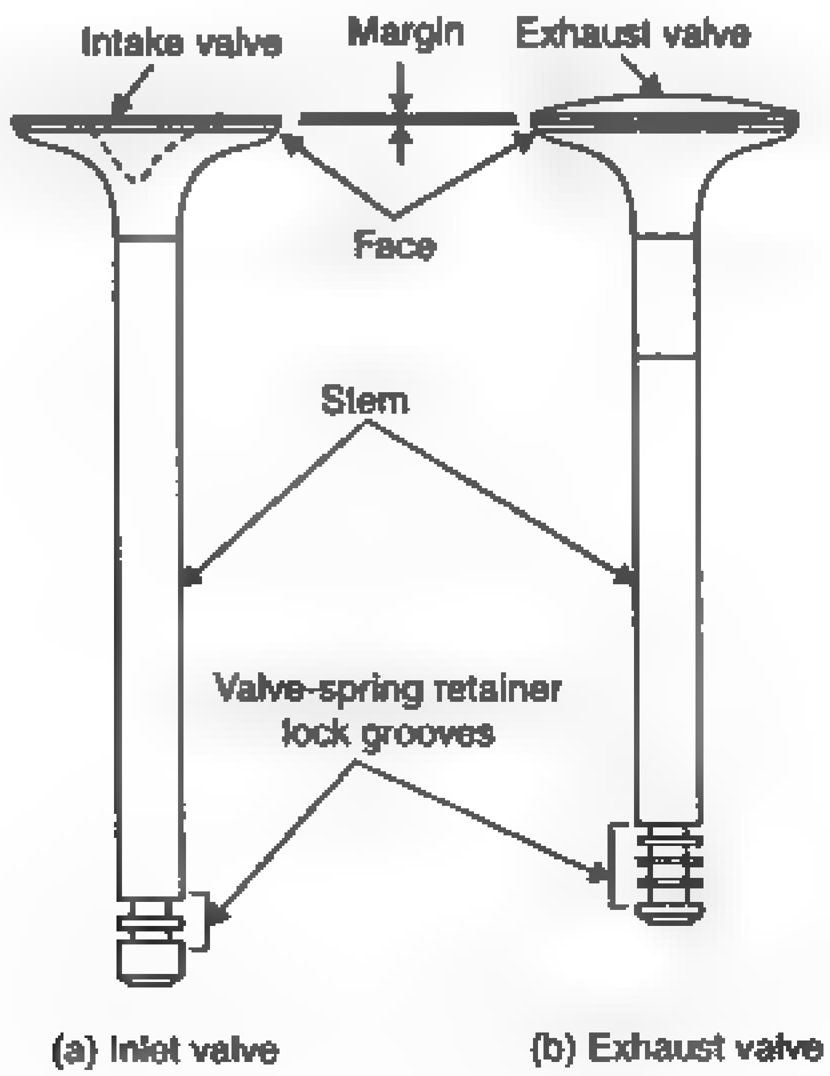


Fig. 2.8

fuel and exhaust valve [Fig. 2.8 (*b*)] for discharging the product of combustion. Two valves are kept closed, by means of cams (Fig. 2.9) geared to the engine shaft. The passage in the cylinder head leading to and from the valves are called *ports*. The pipes which connect the inlet ports of the various cylinders to a common intake pipe for the engine is called the inlet *manifold*. If the exhaust ports are similarly connected to a common exhaust system, this system of piping is called *exhaust manifold*.

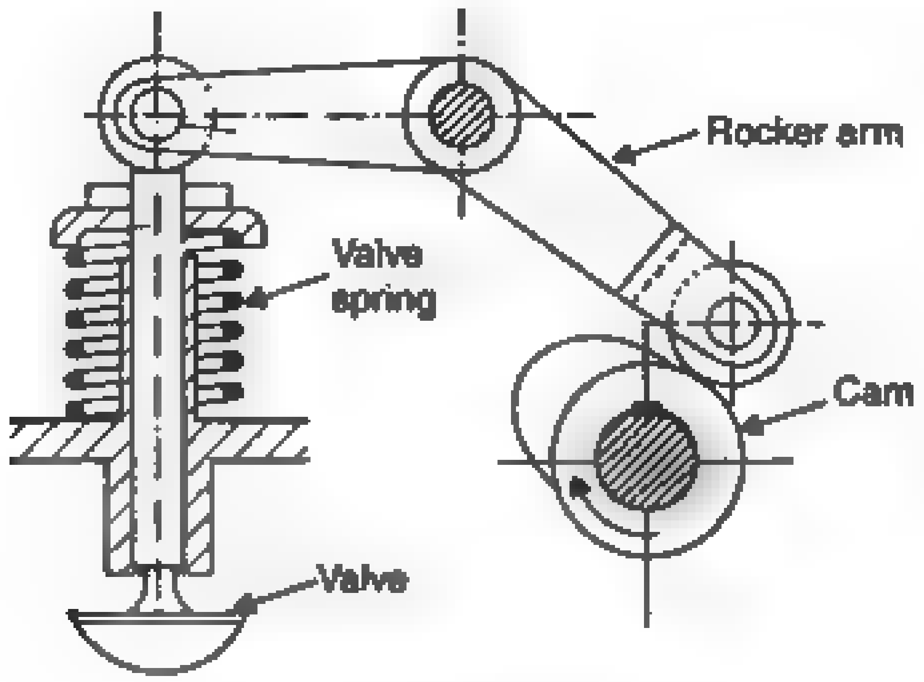


Fig. 2.9. Cam and rocker arm.

The main purpose of the cylinder head is to seal the working ends of the cylinders and not to permit entry and exit of gases on cover head valve engines. The inside cavity of head is called the *combustion chamber*, into which the mixture is compressed for firing. Its *shape controls the direction and rate of combustion*. Heads are drilled and tapped with correct thread to take the ignition spark plug. All the combustion chambers in an engine must be of same shape and size. The shape may be in part controlled by the piston shape.

The cylinder head is usually made of cast iron or aluminium.

3. Piston

A piston is fitted to each cylinder as a face to receive gas pressure and transmit the thrust to the connecting rod.

The piston must (*i*) give gas tight seal to the cylinder through bore, (*ii*) slide freely, (*iii*) be light and (*iv*) be strong. The thrust on the piston on the power stroke tries to tilt the piston as the connecting rod swings, side ways. The piston wall, called the skirt must be strong enough to stand upto this side thrust. *Pistons are made of cast iron or aluminium alloy for lightness*. Light alloy pistons expand more than cast iron one therefore they need large clearances to the bore, when cold, or special provision for expansion. Pistons may be solid skirt or split skirt. A section through a split skirt piston is shown in Fig. 2.10.

4. Piston rings

The piston must be a fairly loose fit in the cylinder. If it were a tight fit, it would expand as it got hot and might stick tight in the cylinder. If a piston sticks it could ruin the engine. On the other hand, if there is too much clearance between the piston and cylinder walls, much of the pressure from the burning gasoline vapour will leak past the piston. This means, that the push on the piston will be much less effective. It is the push on the piston that delivers the power from the engines.

To provide a good sealing fit between the piston and cylinder, pistons are equipped with piston rings, as shown in Fig. 2.10. The rings are usually made of cast iron of fine grain and high elasticity which is not affected by the working heat. Some rings are of alloy spring steel. They are

split at one point so that they can be expanded and slipped over the end of the piston and into ring grooves which have been cut in the piston. When the piston is installed in the cylinder the rings

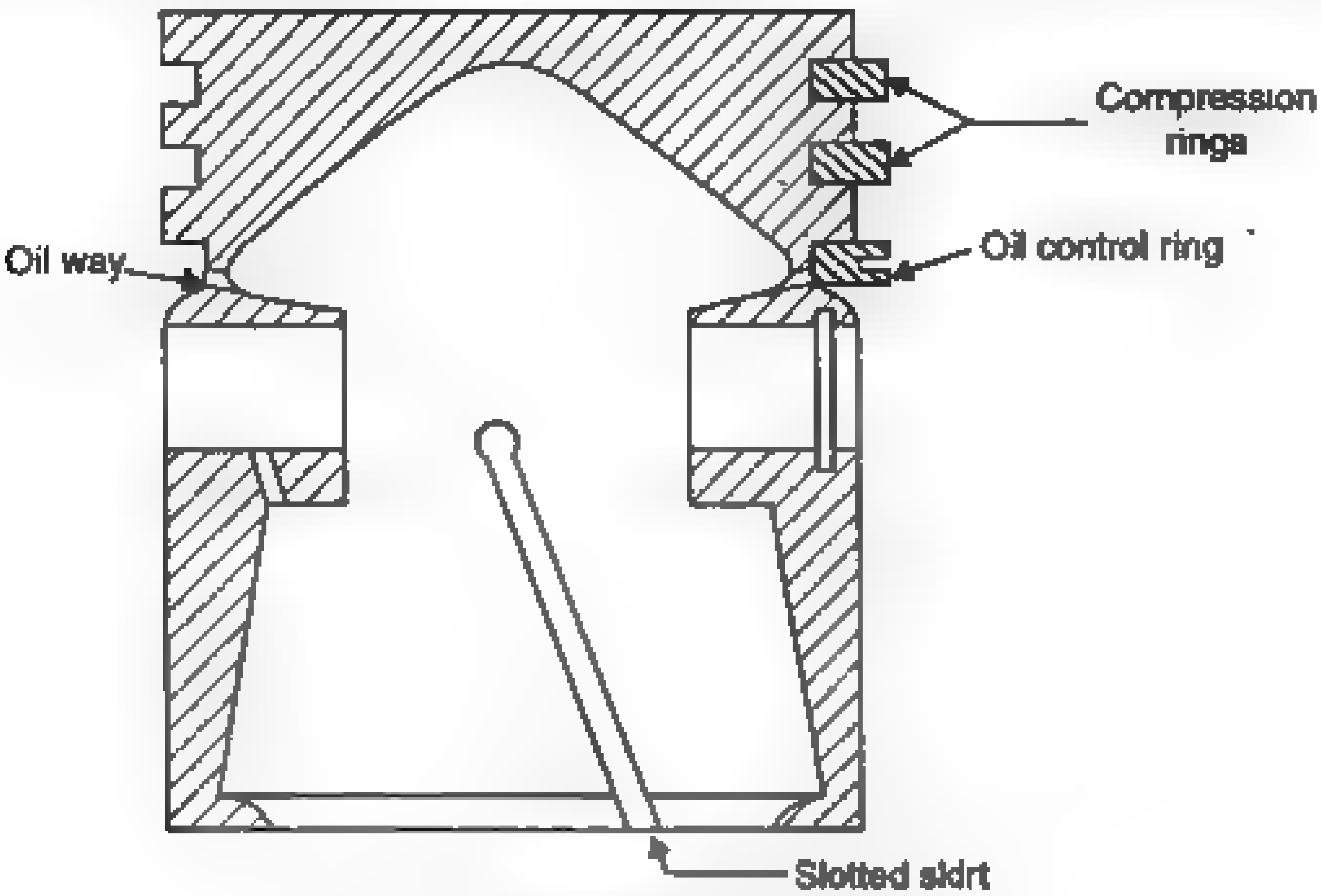


Fig. 2.10. Section through a split skirt piston.

are compressed into ring grooves which have been cut in the piston. When the piston is installed in the cylinder, the rings are compressed into the ring grooves so that the split ends come almost together. The rings fit tightly against the cylinder wall and against the sides of the ring grooves in the piston. Thus, *they form a good seal between the piston and the cylinder wall*. The rings can expand or contact as they heat and cool and still make a good deal. Thus they are free to slide up and down the cylinder wall.

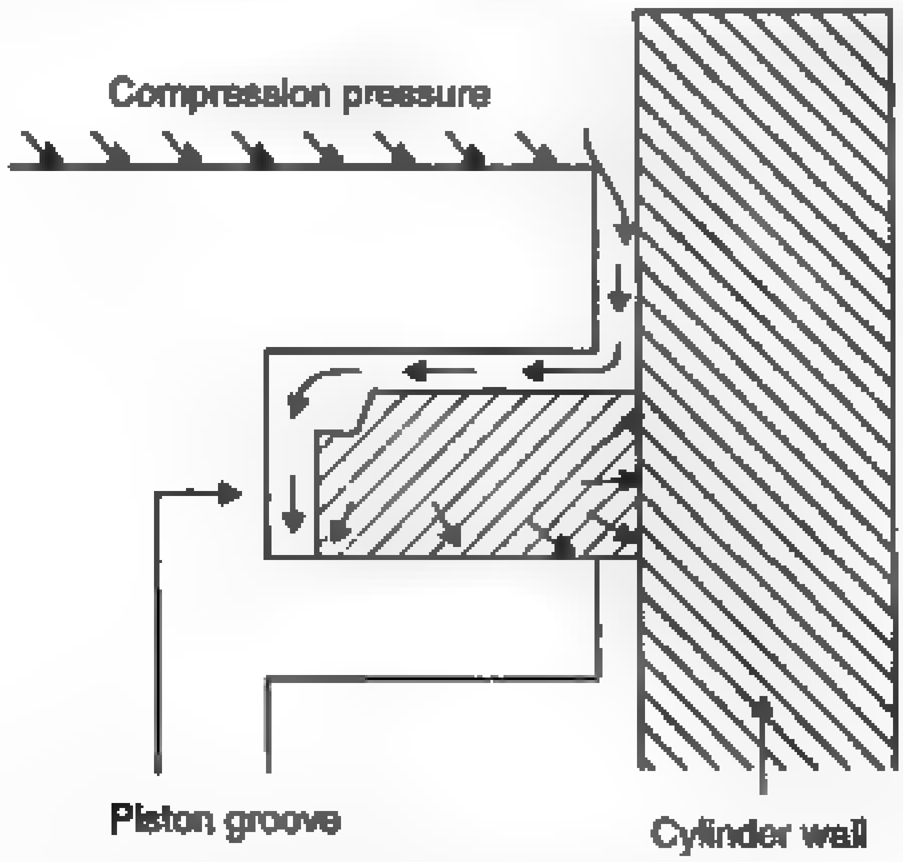


Fig. 2.11. Working of a piston ring.

Fig. 2.11 shows how the piston ring works to hold in the compression and combustion pressure. The arrows show the pressure above the piston passing through clearance between the

piston and the cylinder wall. It presses down against the top and against the back of the piston rings as shown by the arrows. This pushes the piston ring firmly against the bottom of the piston ring groove. As a result there are good seals at both of these points. The higher the pressure in the combustion chamber, the better the seal.

Small two stroke cycle engines have two rings on the piston. Both are compression rings (Fig. 2.12). Two rings are used to divide up the job of holding the compression and combustion pressure. This produces better sealing with less ring pressure against the cylinder wall.

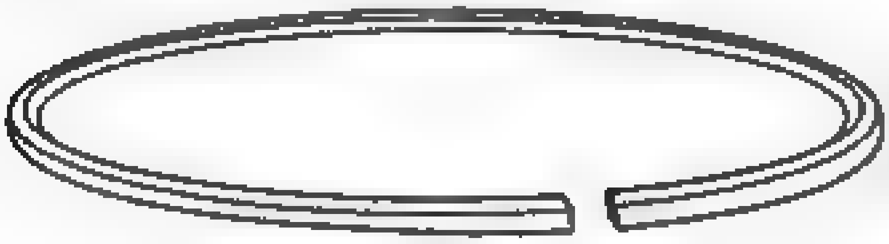


Fig. 2.12. Compression ring.

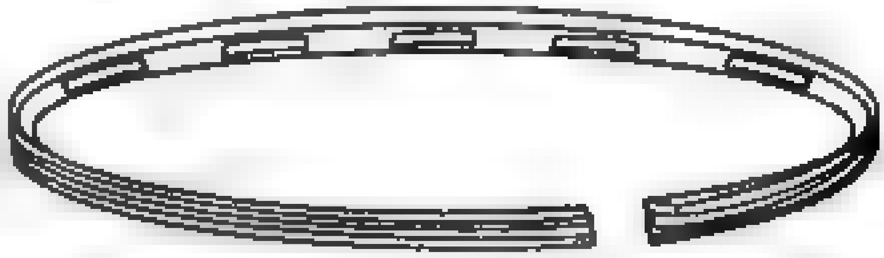


Fig. 2.13. Oil ring.

Four stroke cycle engines have an extra ring, called the oil control ring (Fig. 2.13). Four stroke cycle engines are so constructed that they get much more oil in the cylinder wall than do two stroke cycle engines. This additional oil must be scraped off to prevent it from getting up into the combustion chamber, where it would burn and cause trouble.

Refer Figs. 2.12 and 2.13, the compression rings have a rectilinear cross-section and oil rings are provided with a groove in the middle and with through holes spaced at certain interval from each other. The oil collected from the cylinder walls flows through these holes into the piston groove whence through the holes in the body of the piston and down its inner walls into the engine crankcase.

5. **Gudgeon pin** (or wrist pin or piston pin)

These are *hardened steel parallel spindles* fitted through the piston bosses and the small end bushes or eyes to allow the connecting rods to swivel. Gudgeon pins are a press fit in the piston bosses of light alloy pistons when cold. For removal or fitting, the piston should be dipped in hot water or hot oil, this expands the bosses and the pins can be removed or fitted freely without damage.

It is made hollow for lightness since it is a reciprocating part.

6. **Connecting rod**

Refer Fig. 2.14. The connecting rod transmits the piston load to the crank, causing the latter to turn, thus converting the reciprocating motion of the piston into a rotary motion of the crankshaft. The lower or "big end" of the connecting rod turns on "crank pins".

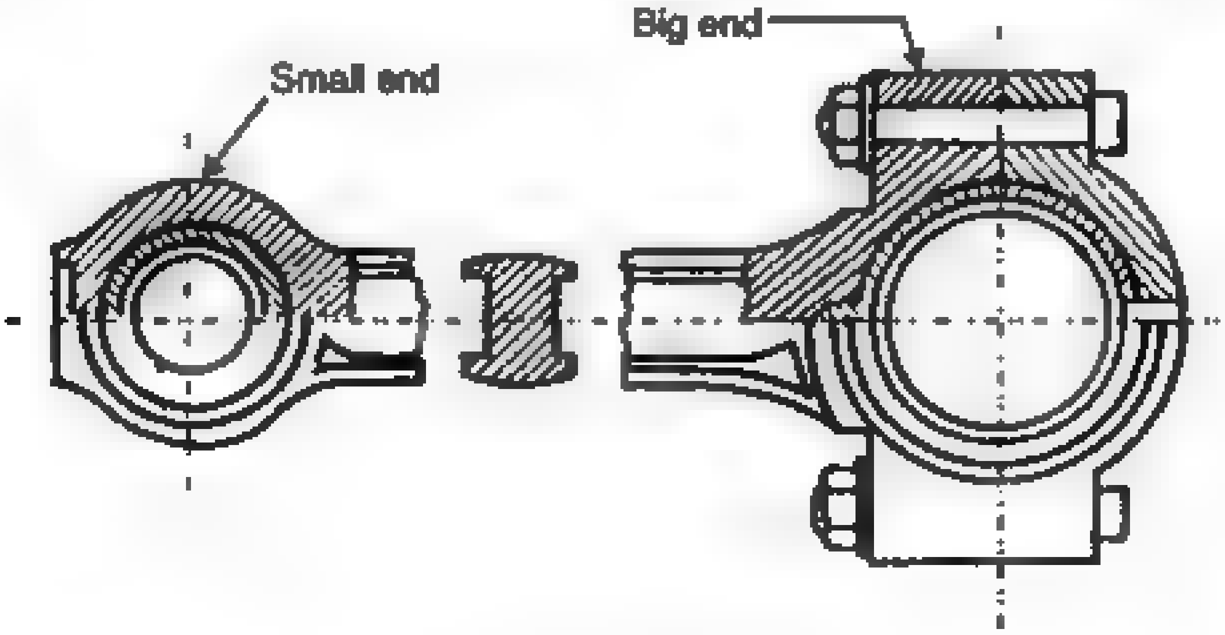


Fig. 2.14. Connecting rod.

The connecting rods are made of nickle, chrome and chrome vandium steels. For small engines the material may be aluminium.

7. Crank

The piston moves up and down in the cylinder. This up and down motion is called *reciprocating motion*. The piston moves in a straight line. The straight line motion must be changed to rotary, or turning motion, in most machines, before it can do any good. That is rotary motion is required to make wheels turn, a cutting blade spin or a pulley rotate. To change the reciprocating motion to rotary motion a crank and connecting rod are used. (Figs. 2.15 and 2.16). The connecting rod connects the piston to the crank.

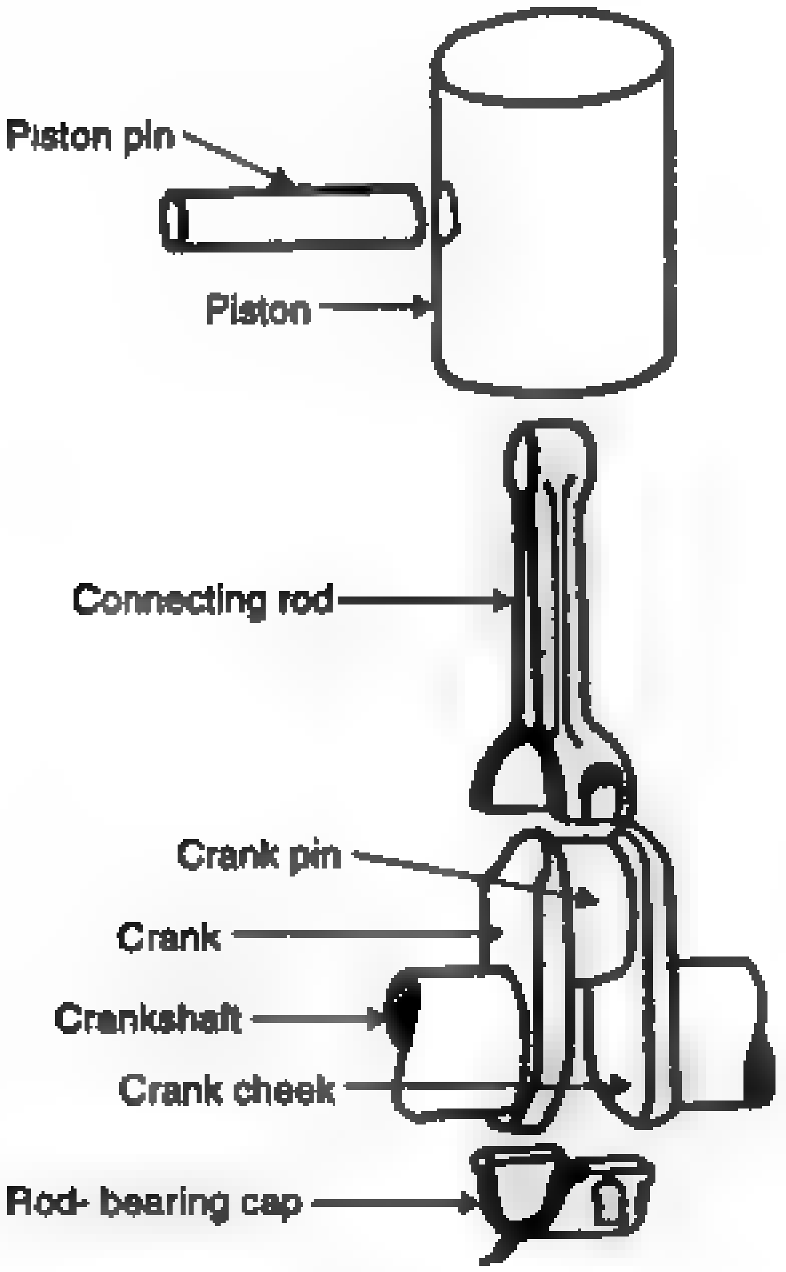


Fig. 2.15

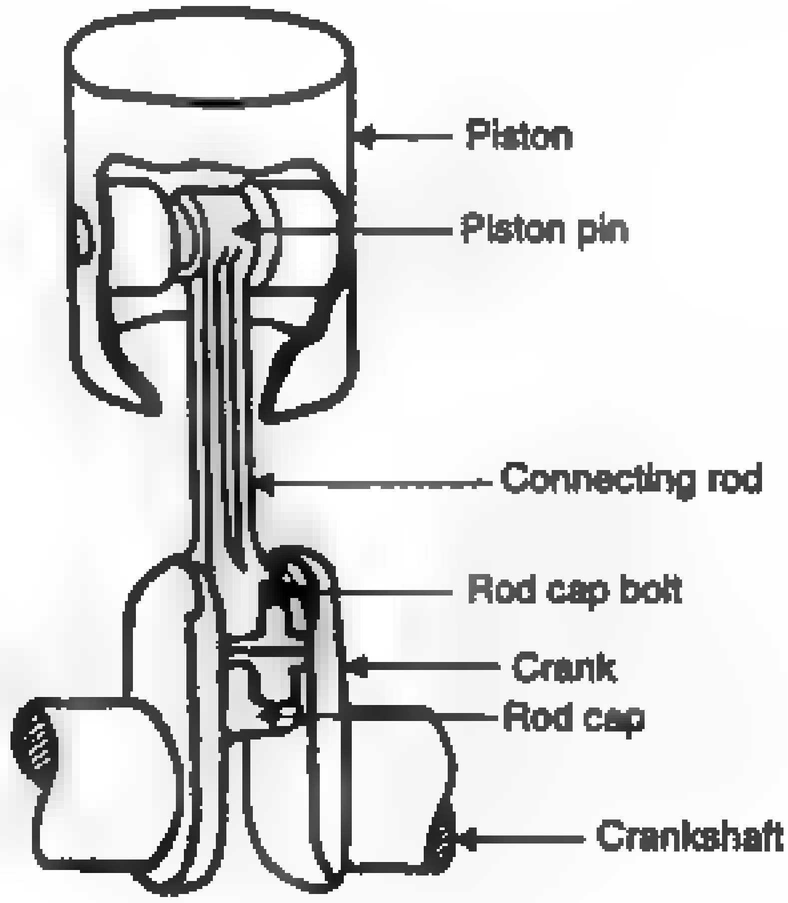


Fig. 2.16

Note. The crank end of the connecting rod is called rod "*big end*". The piston end of the connecting rod is called the rod "*small end*".

8. Crankshaft

The crank is part of the crankshaft. The crankshaft of an internal combustion engine receives via its cranks the efforts supplied by the pistons to the connecting rods. All the engines auxiliary mechanisms with mechanical transmission are geared in one way or the another to the crankshaft. *It is usually a steel forging, but some makers use special types of cast iron such as spheroidal graphitic or nickel alloy castings* which are cheaper to produce and have good service

life. Refer Fig. 2.17. The crankshaft converts the reciprocating motion to rotary motion. The crank shaft mounts in bearings which, encircle the journals so it can rotate freely.

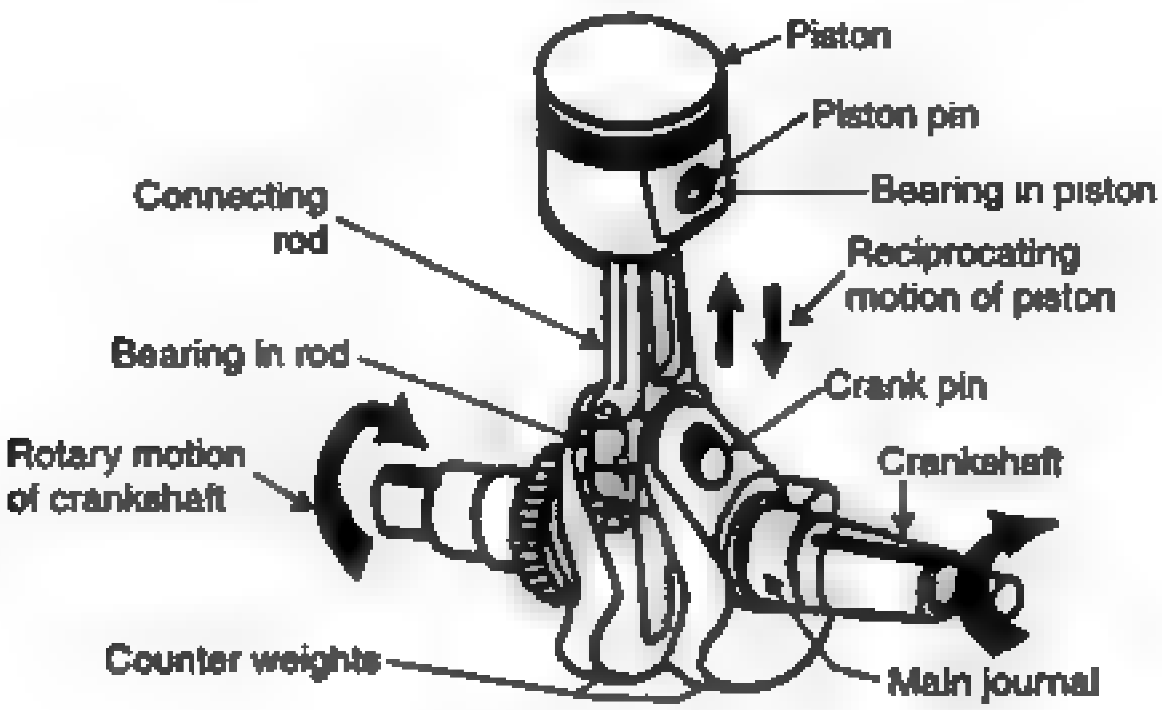


Fig. 2.17. Crank shaft and other parts.

The shape of the crankshaft *i.e.* the mutual arrangement of the cranks depend on the number and arrangement of cylinders and the turning order of the engine. Fig. 2.18 shows a typical crankshaft layout for a four cylinder engine.

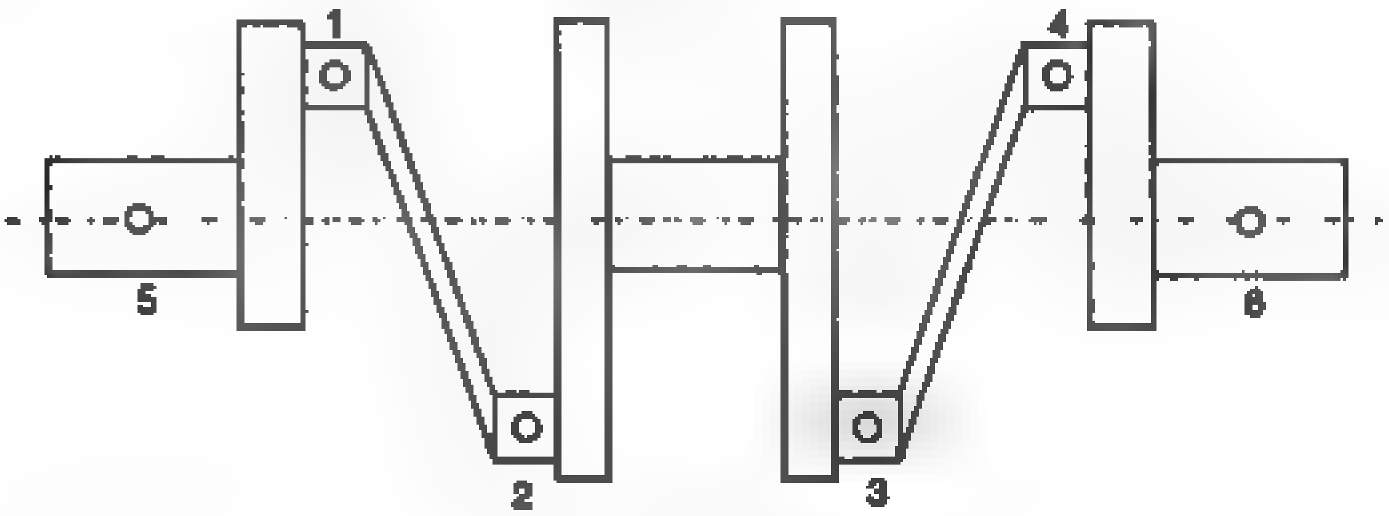


Fig. 2.18. Typical crankshaft layout.

9. Engine bearing

The crankshaft is supported by bearing. The connecting rod big end is attached to the crank pin on the crank of the crankshaft by a bearing. A piston pin at the rod small end is used to attach the rod to the piston. The piston pin rides in bearings. Every where there is rotary action in the engine, bearings are used to support the moving parts. The purpose of bearing is to reduce the friction and allow the parts to move easily. Bearings are lubricated with oil to make the relative motion easier.

Bearings used in engines are of two types : *sliding* or *rolling* (Fig. 2.19).

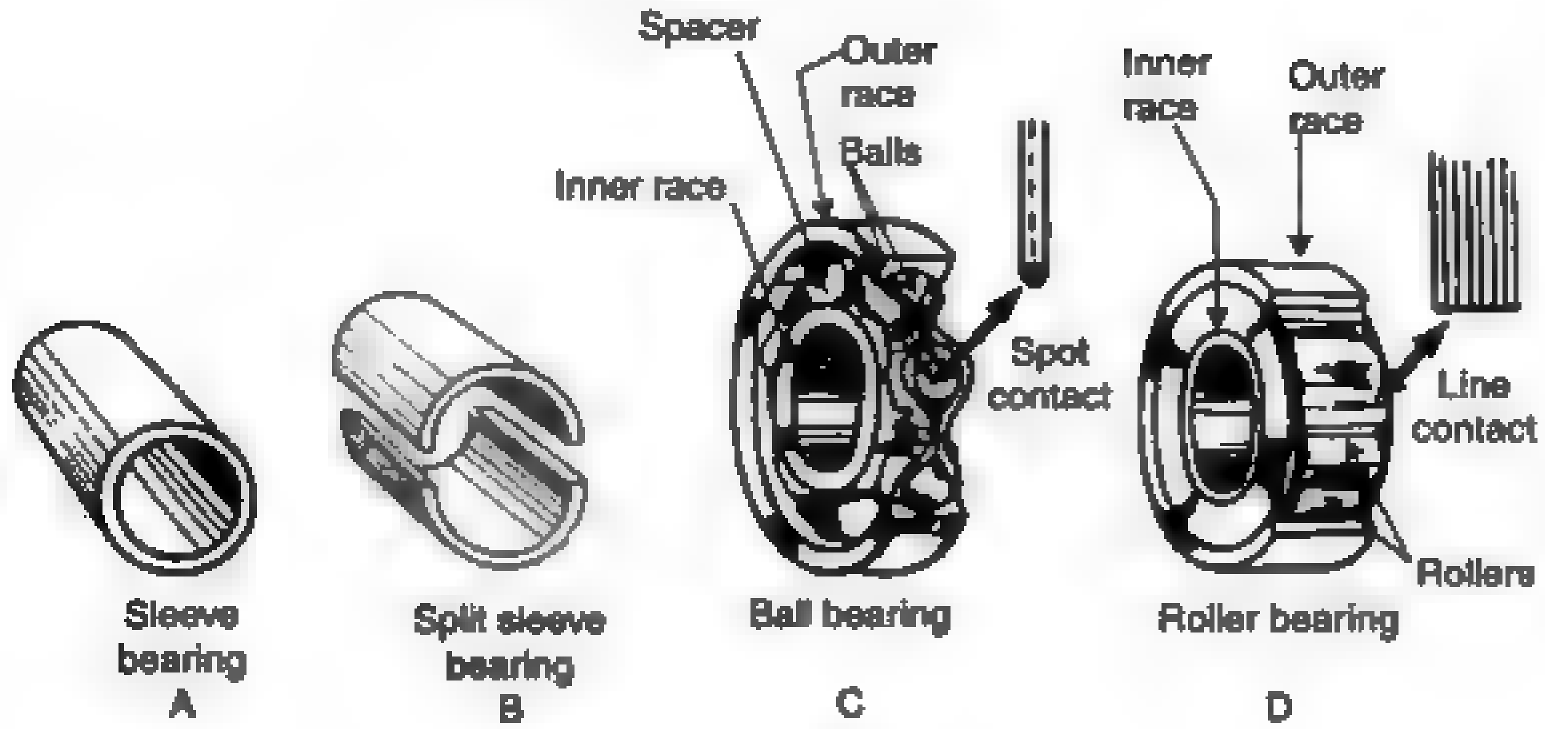


Fig. 2.19. Bearings.

The sliding type of bearings are sometimes called *bushings or sleeve bearings* because they are in the shape of a sleeve that fits around the rotating journal or shaft. The sleeve-type connecting rod big end bearings usually called simply rod bearings and the crankshaft supporting bearings called the main bearings are of the split sleeve type. They must be split in order to permit their assembly into the engine. In the rod bearing, the upper half of the bearing is installed in the rod, the lower half is installed in the rod bearing cap. When the rod cap is fastened to the rod shown in Fig. 2.16 a complete sleeve bearing is formed. Likewise, the upper halves of the main bearings are assembled in the engine and then the main bearing caps, with the lower bearing halves are attached to the engine to complete the sleeve bearings supporting the crankshaft.

The typical bearing half is made of steel or bronze back to which a lining of relatively soft bearing material is applied. Refer Fig. 2.20. This relatively soft bearing material, which is made of several materials such as copper, lead, tin and other metals, has the ability to conform to slight irregularities of the shaft rotating against it. If wear does take place, it is the bearing that wears and the bearing can be replaced instead of much more expansive crankshaft or other engine part.

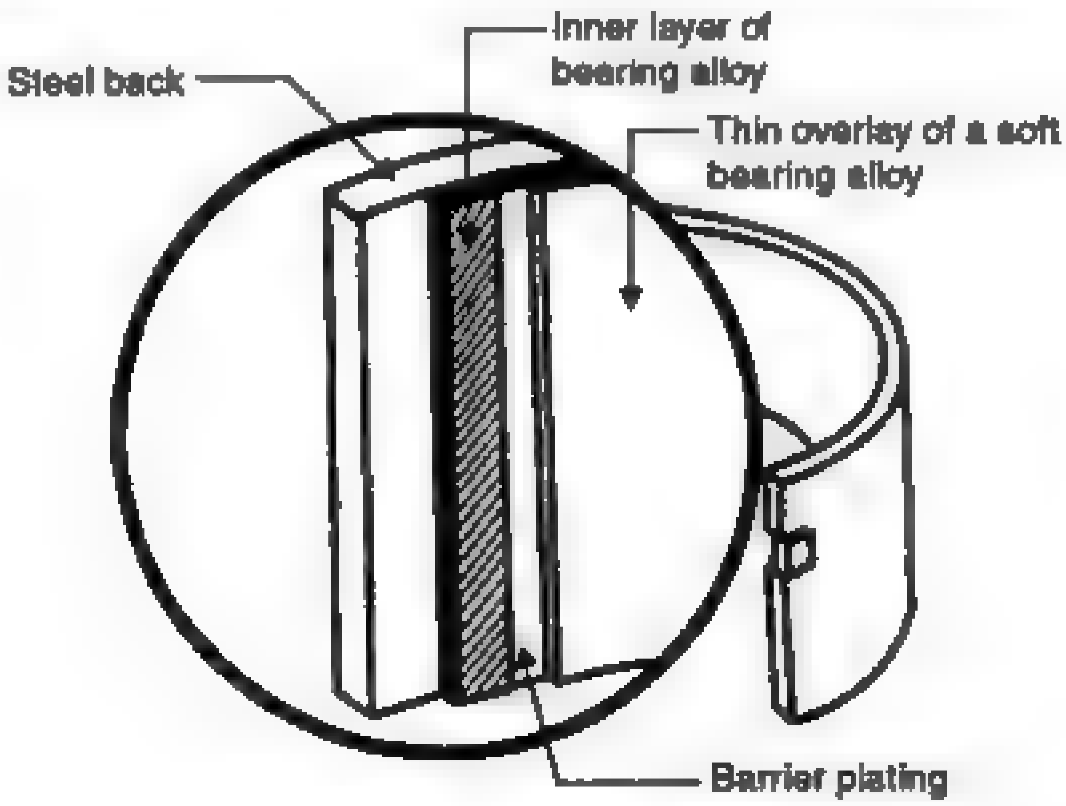


Fig. 2.20. Bearing half (details).

The rolling-type bearing uses balls or rollers between the stationary support and the rotating shaft. Refers Fig. 2.19. Since the balls or rollers provide rolling contact, the frictional resistance to movement is much less. In some roller bearing, the rollers are so small that they are hardly bigger than needles. These bearings are called *needle bearings*. Also some rollers bearings have the rollers set at an angle to the races, the rollers roll in are tapered. These bearings are called *tapered roller bearings*. Some ball and roller bearings are sealed with their lubricant already in place. Such bearings require no other lubrication. Other do require lubrication from the oil in the gasoline (two stroke cycle engines) or from the engine lubrication system (four stroke cycle engines).

The type of bearing selected by the designers of the engine depends on the design of the engine and the use to which the engine will be put. *Generally, sleeve bearings, being less expensive and satisfactory for most engine applications, are used. In fact sleeve bearings are used almost universally in automobile engines. But you will find some engines with ball and roller bearings to support the crankshaft and for the connecting rod and piston-pin bearings.*

10. **Crankcase**

The main body of the engine to which the cylinders are attached and which contains the crankshaft and crankshaft bearing is called *crankcase*. This member also holds other parts in alignment and resists the explosion and inertia forces. It also protects the parts from dirt etc. and serves as a part of lubricating system.

11. **Flywheel**

Refer Figs. 2.4 and 2.21. A flywheel (steel or cast iron disc) secured on the crank shaft performs the following *functions* :

(*a*) Brings the mechanism out of dead centres.

(*b*) Stores energy required to rotate the shaft during preparatory strokes.

(*c*) Makes crankshaft rotation more uniform.

(*d*) Facilitates the starting of the engine and overcoming of short time over loads as, for example, when the machine is started from rest.

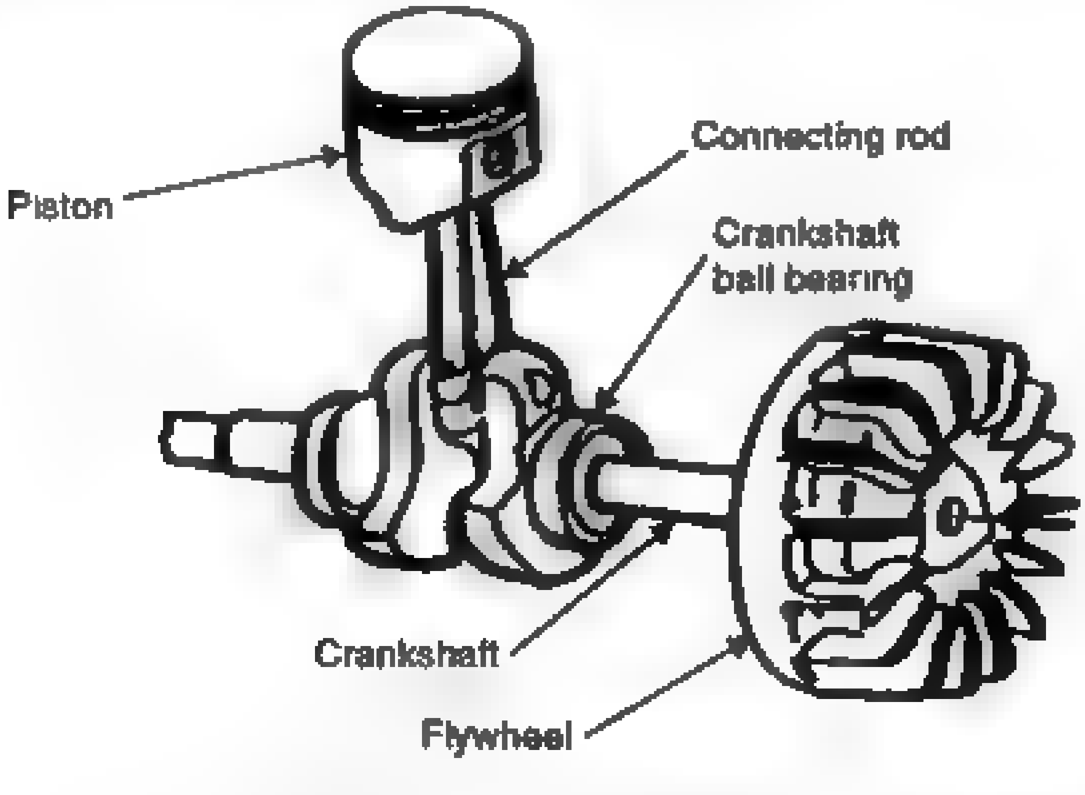


Fig. 2.21. Flywheel secured on crankshaft.

The weight of the flywheel depends upon the nature of variation of the pressure. The flywheel for a double-acting steam engine is lighter than that of a single-acting one. Similarly, the flywheel for a two-stroke cycle engine is lighter than a flywheel used for a four-stroke cycle engine. *Lighter flywheels are used for multi-cylinder engines.*

12. Governor

A governor may be defined *as a device for regulating automatically output of a machine by regulating the supply of working fluid.* When the speed decreases due to increase in load the supply valve is opened by mechanism operated by the governor and the engine therefore speeds up again to its original speed. If the speed increases due to a decrease of load the governor mechanism closes the supply valve sufficiently to slow the engine to its original speed. *Thus the function of a governor is to control the fluctuations of engine speed due to changes of load.*

Comparison between a Flywheel and a Governor

	Flywheel	*Governor*
1.	It is provided on engines and fabricating machines *viz.*, rolling mills, punching machines ; shear machines, presses etc.	It is provided on prime movers such as engines and turbines.
2.	Its function is to store the available mechanical energy when it is in excess of the load requirement and to part with the same when the available energy is less than that required by the load.	Its function is to regulate the supply of driving fluid producing energy, according to the load requirement so that at different loads almost a constant speed is maintained.
3.	It works continuously from cycle to cycle.	It works intermittently *i.e.* only when there is change in load.
4.	In engines it takes care of fluctuations of speed during thermodynamic cycle.	It takes care of fluctuations of speed due to variation of load over long range of working engines and turbines.
5.	In fabrication machines it is very economical to use it in that it reduces capital investment on prime movers and their running expenses.	But for governor, there would have been unnecessarily more consumption of driving fluid. Thus it economies its consumption.

Types of governor :

Governors are classified as follows :

1. Centrifugal governor

(*i*) *Gravity controlled*, in which the centrifugal force due to the revolving masses is largely balanced by gravity.

(*ii*) *Spring controlled*, in which the centrifugal force is largely balanced by springs.

2. Inertia and flywheel governors

(*i*) Centrifugal type, in which centrifugal forces play the major part in the regulating action.

(*ii*) Inertia governor, in which the inertia effect predominates.

The *inertia type* governors are fitted to the crankshaft or flywheel of an engine and so differ radically in appearance from the centrifugal governors. The balls are so arranged that the inertia force caused by an angular acceleration or retardation of the shaft tends to alter their positions. The amount of displacement of governor balls is controlled by suitable springs and through the governor mechanism, alters the fuel supply to the engine. The inertia governor is more sensitive than centrifugal but it becomes very difficult to balance the revolving parts. For this reason *centrifugal governors are more frequently used.* We shall discuss centrifugal governors only.

Important centrifugal governors are :

1. Watt governor
2. Porter governor
3. Proell governor
4. Hartnell governor.

1. Watt governor

It is the primitive governor as used by Watt on some of his early steam engines. It is used for a very slow speed engine and this is why it has now become obsolete.

Refer Fig. 2.22. Two arms are hinged at the top of the spindle and two revolving balls are fitted on the other ends of the arms. One end of each of the links are hinged with the arms, while the other ends are hinged with the sleeve, which may slide over the spindle. The speed of the crankshaft is transmitted to the spindle through a pair of bevel gears by means of a suitable arrangement. So the rotation of the spindle of the governor causes the weights to move away from the centre due to the centrifugal force. This makes the sleeve to move in the upward direction. This movement of the sleeve is transmitted by the lever to the throttle valve which partially closes or opens the steam pipe and reduces or increases the supply of steam to the engine. So the engine speed may be adjusted to a normal limit.

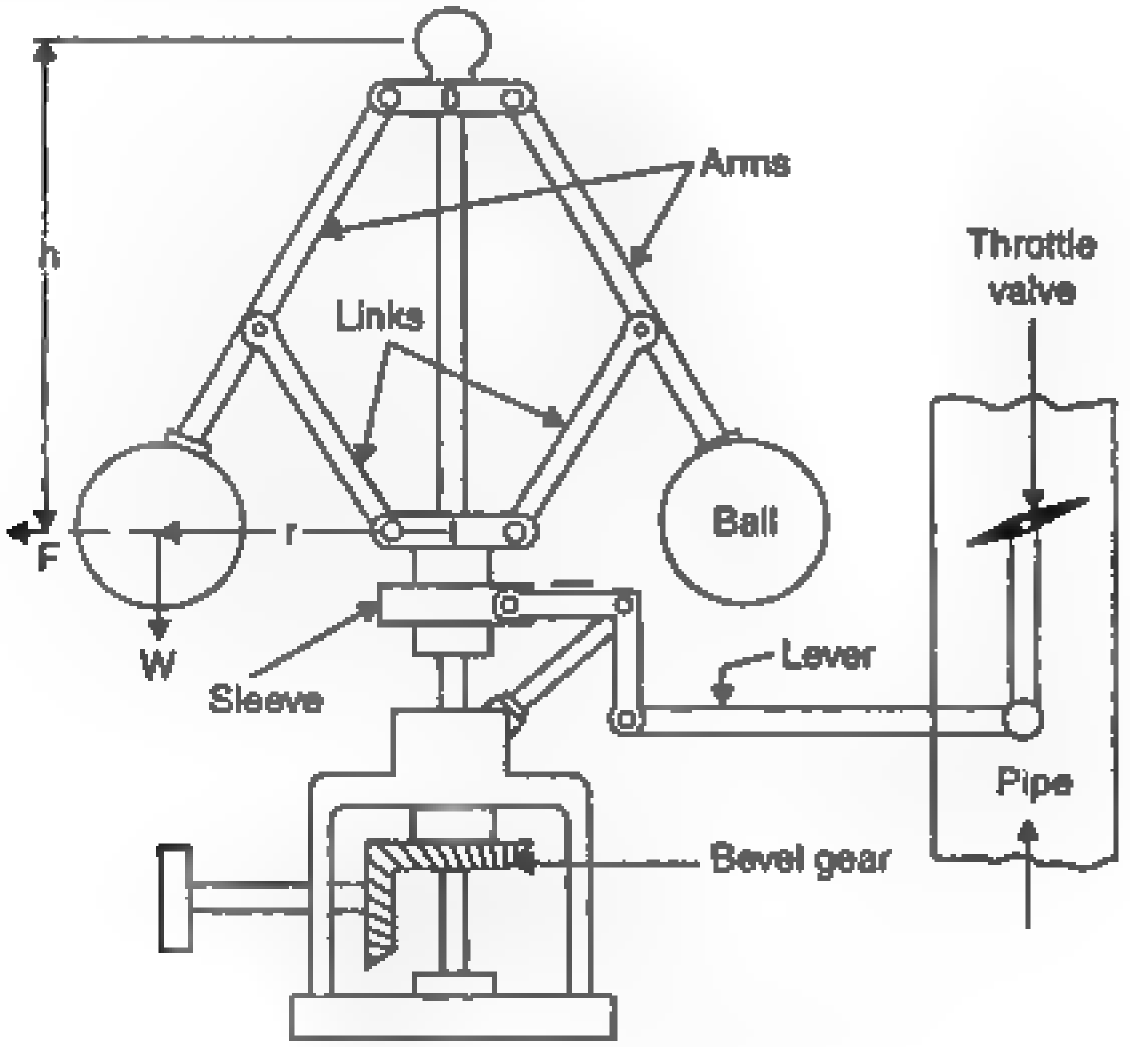


Fig. 2.22. Watt governor.

2. Porter governor

Fig. 2.23 shows diagrammatically a Porter governor where two or more masses called the governor balls rotate about the axis of the governor shaft which is driven through suitable gearing from the engine crankshaft. The governor balls are attached to the arms. The lower arms are attached to the *sleeve which acts as a central weight*. If the speed of the rotation of the balls increases owing to a decrease of load on the engine, the governor balls fly outwards and the sleeve moves upwards thus closing the fuel passage till the engine speed comes back to its designed speed. If the engine speed decreases owing to an increase of load, the governor balls fly inwards and the sleeve moves downwards thus opening the fuel passage more for oil till the engine speed comes back to its designed speed. The engine is said to be running at its designed speed when the outward inertia or centrifugal force is just balanced by the inward controlling force.

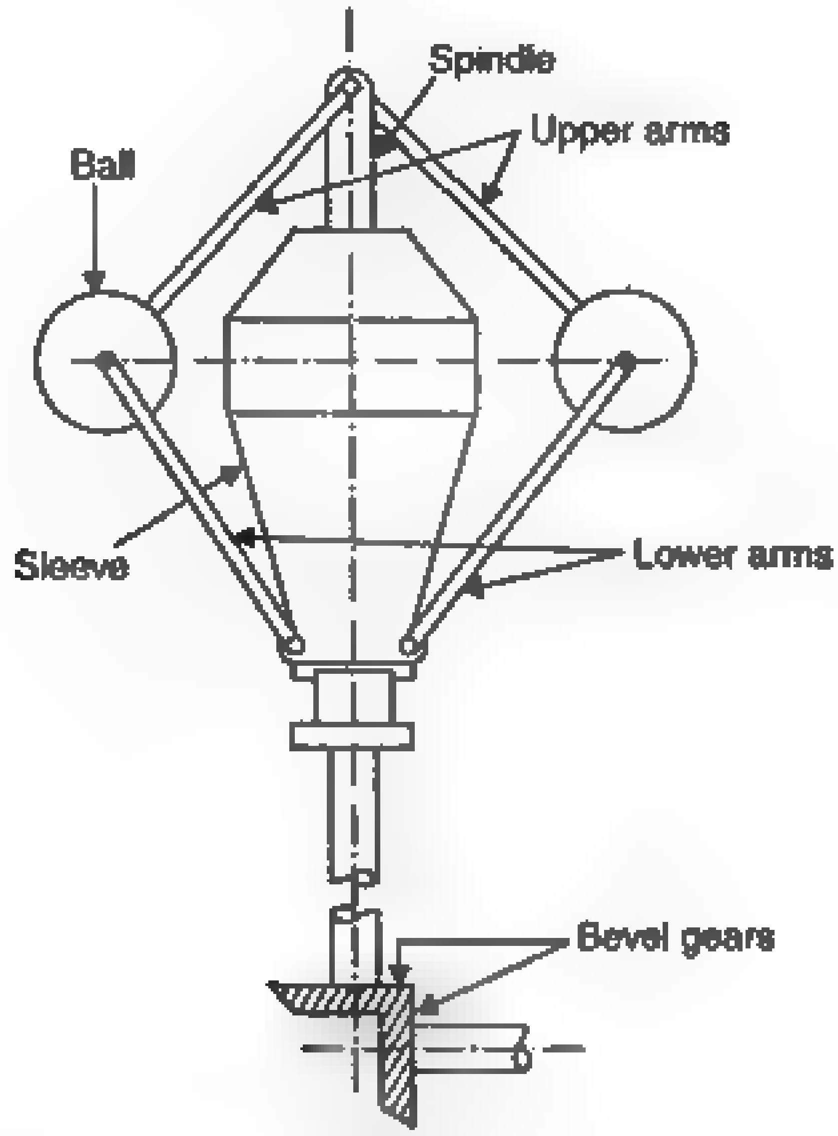


Fig. 2.23. Porter governor.

3. Proell governor

Refer Fig. 2.24. It is a modification of porter governor. The governor balls are carried on an *extension of the lower arms*. For given values of weight of the ball, weight of the sleeve and height

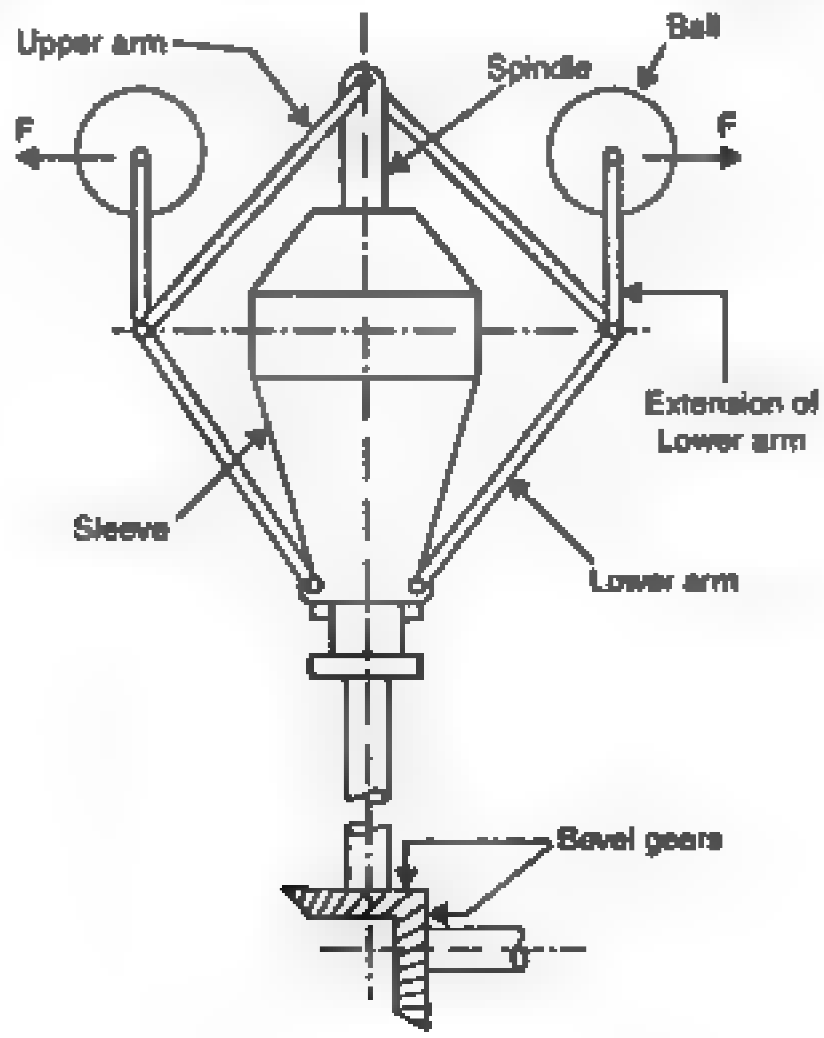


Fig. 2.24. Proell governor

of the governor, a Proell governor runs at a *lower speed* than a Porter governor. *In order to give the same equilibrium speed a ball of smaller mass may be used in Proell governor.*

4. Hartnell governor

The Hartnell governor is a spring loaded governor in which the controlling force, to a great extent, is provided by the spring thrust.

Fig. 2 25 shows one of the types of Hartnell governors. It consists of casing fixed to the spindle. A compressed spring is placed inside the casing which presses against the top of the casing and on adjustable collars. The sleeve can move up and down on the vertical spindle depending upon the speed of the governor. Governor balls are carried on bell crank lever which are pivoted on the lower end of the casing The balls will fly outwards or inwards as the speed of the governor shaft increases or decreases respectively

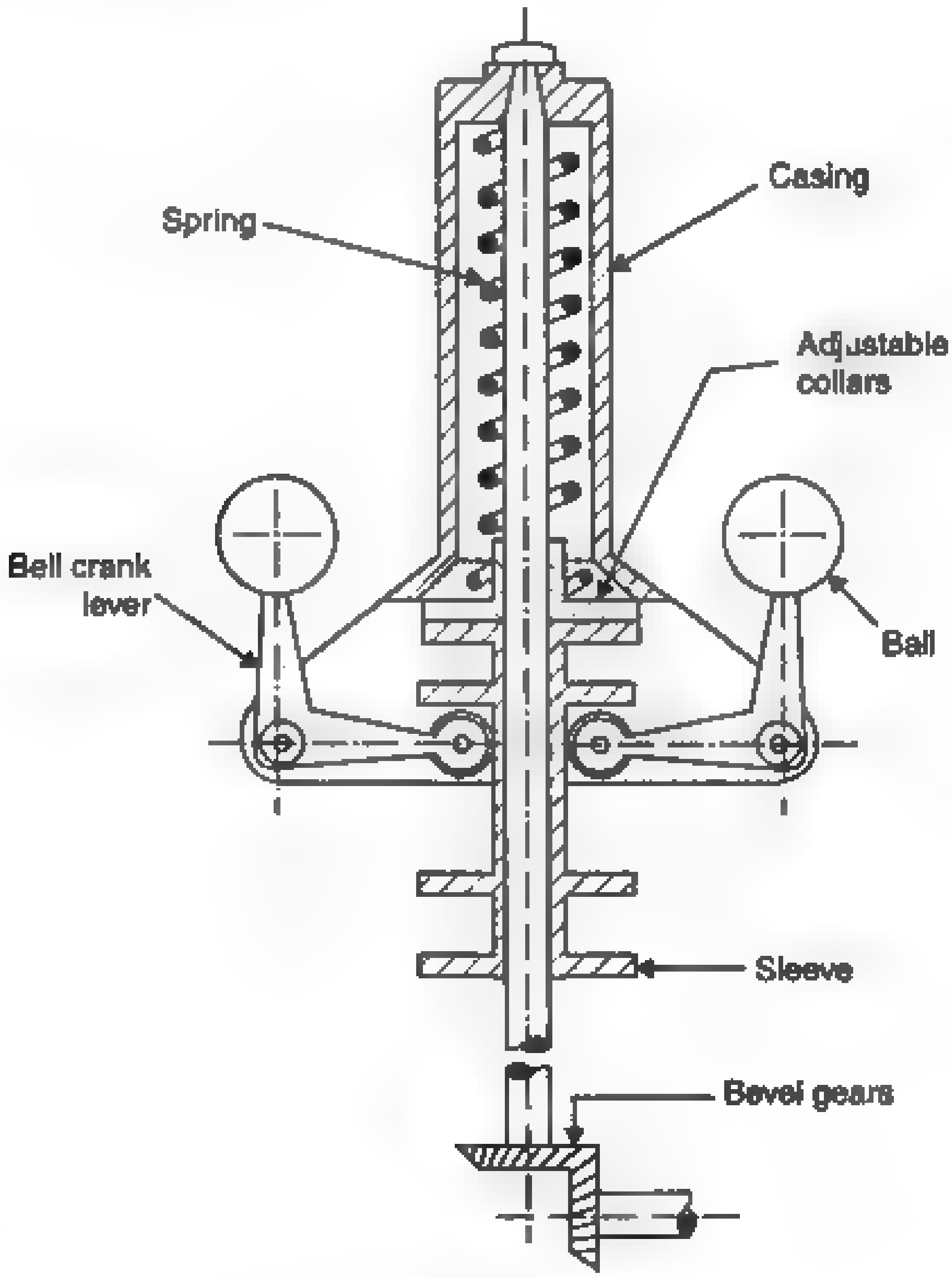


Fig. 2.25. Hartnell gbovernor

5. Valves and valve gears

With few exceptions the inlet and exhaust of internal combustion engines are controlled by poppet valves. These valves are held to their seating by strong springs, and as the valves usually

open inwards, the pressure in the cylinder helps to keep them closed. The valves are lifted from their seats and the ports opened either by cams having projecting portion designed to give the period of opening required or by eccentrics operating through link-work. Of these two methods the cam gear is more commonly used, but in either case it is necessary that the valve gear shaft of an engine should rotate but once from beginning to end of a complete cycle, however many strokes may be involved in the completion of that cycle. This is necessary to secure a continuous regulation of the valve gear as required. For this purpose the cams or eccentrics of four-stroke engines are mounted on shafts driven by gearing at half the speed of the crankshaft. The curves used for the acting faces of the cams depend on the speed of the engine and rapidity of valve opening desired.

Fig. 2.26 shows a valve gear for I.C. engine. It consists of poppet valve, the steam bushing or guide, valve spring, spring retainer, lifter or push rod, camshaft and half speed gear for a four-

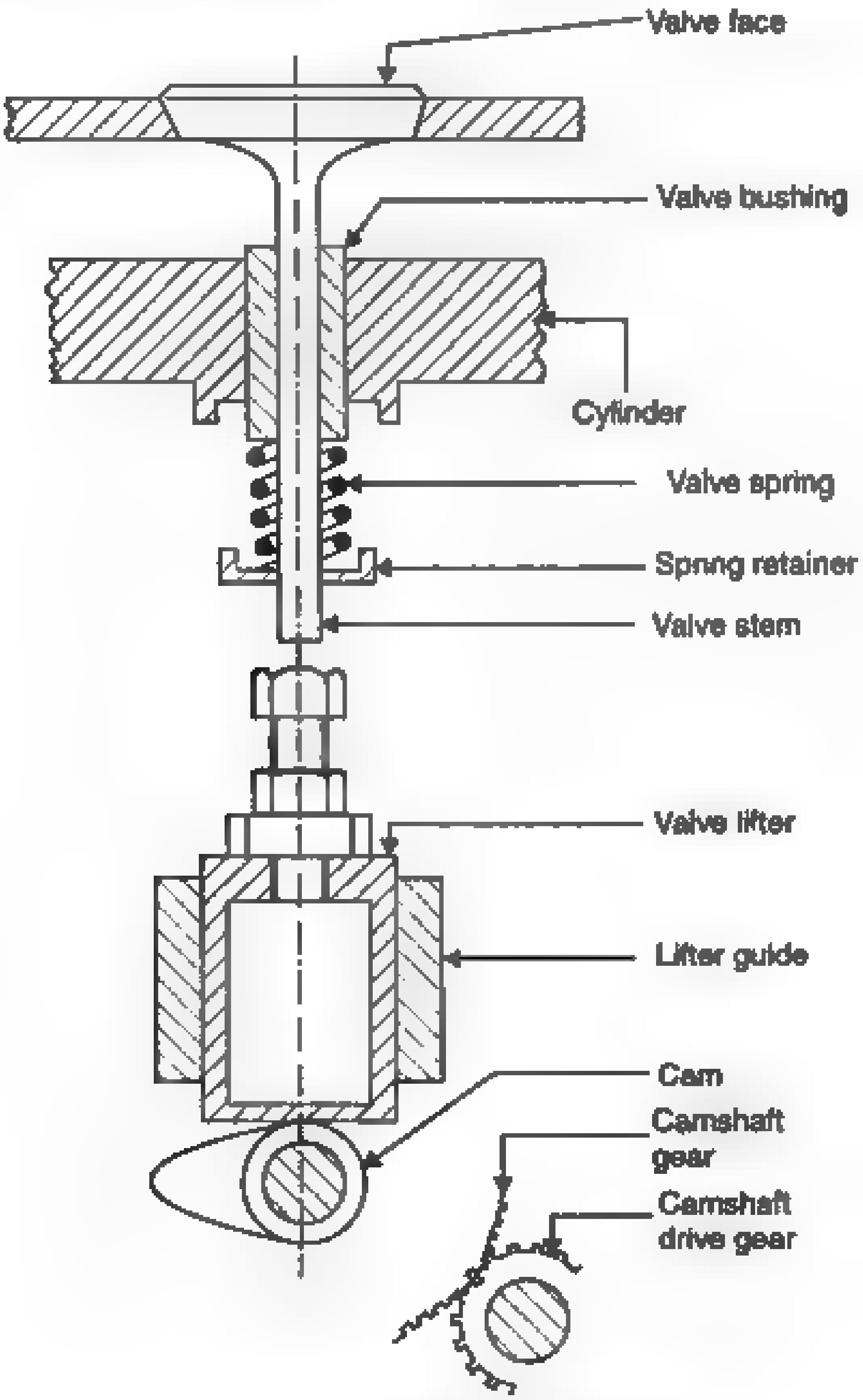


Fig. 2.26. Valve gear for I.C. engine.

stroke engine. The poppet valve, in spite of its shortcomings of noise and difficulties of cooling is commonly used due to its simplicity and capacity for effective sealing under all operating conditions. The valve is subjected to very heavy duty. It holds in combustion chamber and is exposed to high temperatures of burning gases. Exhaust valve itself may attain a high temperature while external cooling is not available. Special heat resisting alloys are therefore used in the construction of the exhaust valve and it may sometimes have a hollow construction filled with mineral salts to provide for heat dissipation. The salts become liquid when valve is working and transfer heat from the head to the stem from which it is carried through the stem guide to the cylinder block.

The timing of the valves *i.e.* their opening and closing with respect to the travel of the piston is very important thing for efficient working of the engine. The drive of the camshaft is arranged through gears or chain and sprocket (called timing gear or timing chain). Any wearing of the gears or chain and sprocket would result in disturbing the precise timing of the valves. It is desirable, therefore, to avoid use of multiple gears of long chains in the camshaft drive.

Valve timing

Theoretically the valves open and close at top dead centre (T D.C.) or at bottom dead centre (B.D.C.) but practically they do so some time before or after the piston reaches the upper or lower limit of travel. There is a reason for this. Look at the inlet valve, for example. It normally opens several degrees of crankshaft-rotation before T.D.C. on the exhaust stroke. That is the intake valve begins to open before the exhaust stroke is finished. This gives the valve enough time to reach the fully open position before the intake stroke begins. Then, when the intake stroke starts, the intake valve is already wide open and air fuel mixture can start to enter the cylinder, immediately. Likewise the intake valve remains open for quite a few degrees of crankshaft rotation after the piston has passed B.D.C. at the end of the intake stroke. This allows additional time for air fuel mixture to continue to flow into the cylinder. The fact that the piston has already passed B.D.C. and is moving up or the compression stroke while the intake valve is still open does not effect the movement of air fuel mixture into the cylinder. Actually air fuel mixture is still flowing in as the intake valve starts to close.

This is due to the fact that air-fuel mixture has inertia. That is, it attempts to keep on flowing after it once starts through the carburettor and into the engine cylinder. The momentum of the mixture then keeps it flowing into the cylinder even though the piston has started up on the compression stroke. This packs more air-fuel mixture into the cylinder and results in a stronger power stroke. In other words, this improves *volumetric efficiency*.

For a some what similar reason, the exhaust valve opens well before the piston reaches B.D.C. on the power stroke. As the piston nears B.D C., most of the push on the piston has ended and nothing is lost by opening the exhaust valve towards the end of the power stroke. This gives the exhaust gases additional time to start leaving the cylinder so that exhaust is well started by the time the piston passes B.D.C. and starts up on the exhaust stroke. The exhaust valve then starts opening for some degrees of crankshaft rotation after the piston has passed T.D.C. and intake stroke has started. This makes good use of momentum of exhaust gases. They are moving rapidly towards the exhaust port, and leaving the exhaust valve open for a few degrees after the intake stroke starts giving the exhaust gases some additional time to leave the cylinder. This allows more air-fuel mixture to enter on the intake stroke so that the stronger power stroke results. That is, it improves volumetric efficiency.

The actual timing of the valves varies with different four stroke cycle engines, but the typical example for an engine is shown in Fig. 2.27. Note that the inlet valve opens 15° of crank-

shaft rotation before T.D.C. on the exhaust stroke and stays open until 50° of crankshaft rotation after B.D.C. on the compression stroke. The exhaust valve opens 50° before B.D.C. on the power stroke and stays open 15° after T.D.C. on the inlet stroke. This gives the two valves an overlap of 30° at the end of exhaust stroke and beginning of the *compression stroke.*

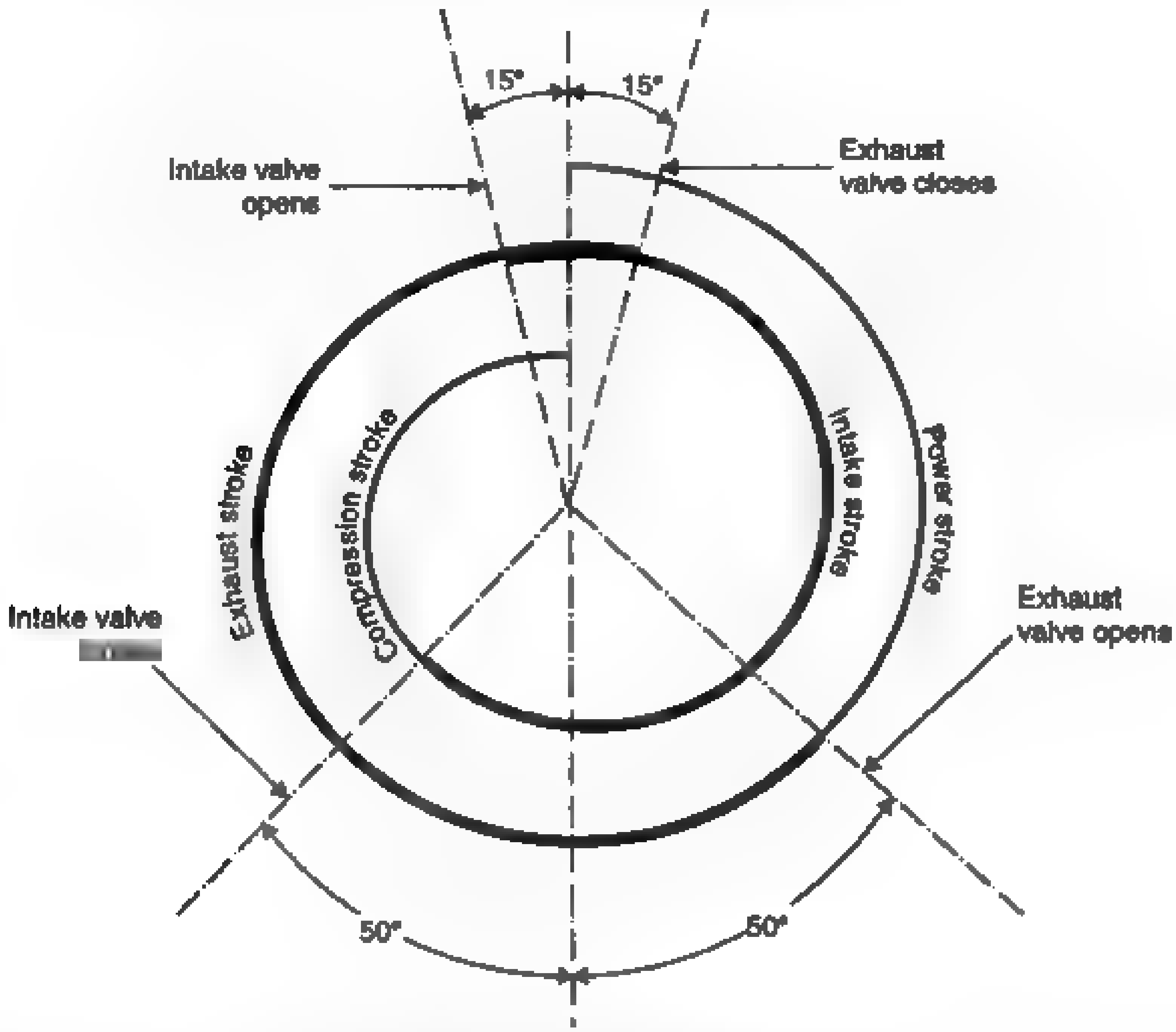


Fig. 2.27. Typical valve timing diagram.

B. Parts common to petrol engine only :

Spark-plug

The main function of a spark-plug is to conduct the high potential from the ignition system into the combustion chamber. It provides the proper gap across which spark is produced by applying high voltage, to ignite the combustion chamber.

A spark-plug entails the following requirements :

(*i*) It must withstand peak pressures up to atleast 55 bar.

(*ii*) It must provide suitable insulation between two electrodes to prevent short circuiting

(*iii*) It must be capable of withstanding high temperatures to the tune of 2000°C to 2500°C over long periods of operation.

(*iv*) It must offer maximum resistance to erosion burning away of the spark points irrespective of the nature of fuel used.

(*v*) It must possess a high heat resistance so that the electrodes do not become sufficiently hot to cause the preignition of the charge within the engine cylinder.

(*vi*) The insulating material must withstand satisfactorily the chemical reaction effects of the fuel and hot products of combustion.

(*vii*) Gas tight joints between the insulator and metal parts are essential under all operating conditions.

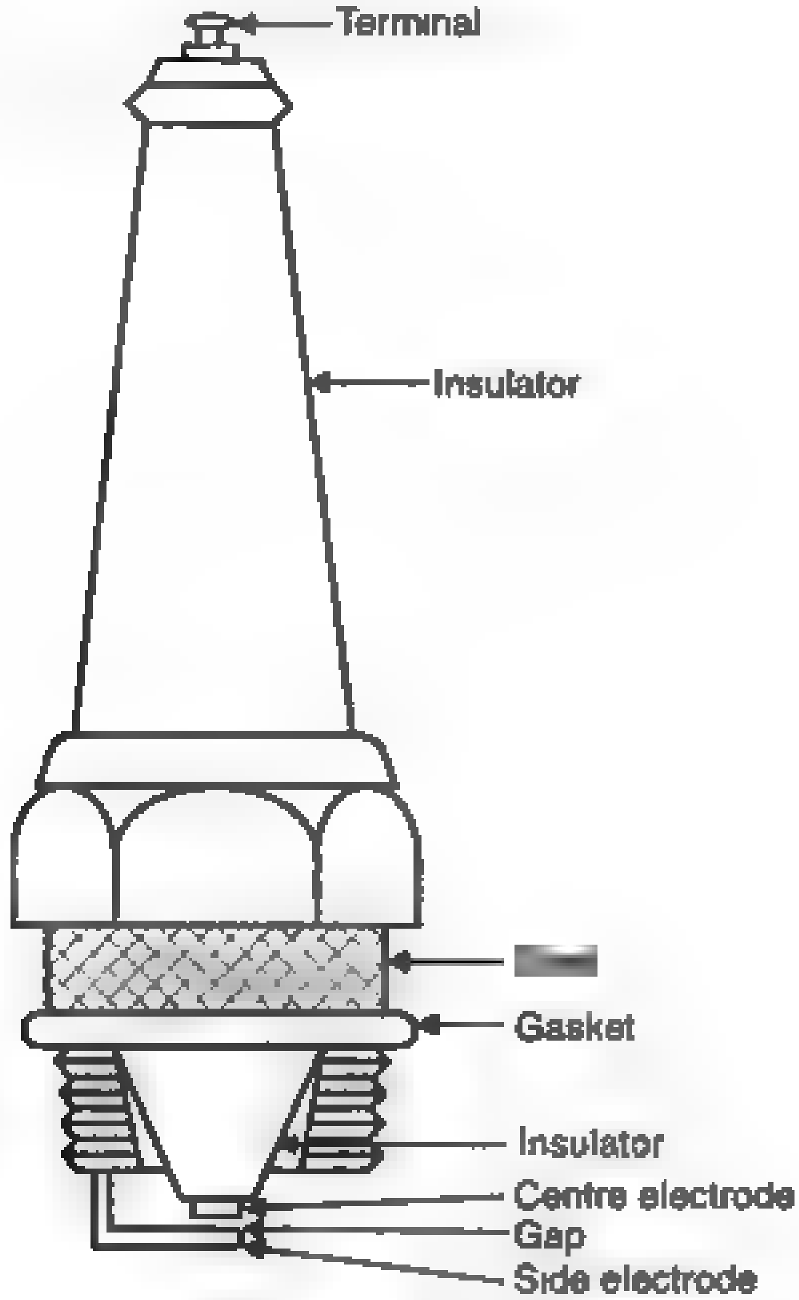


Fig. 2.28. Spark-plug.

Refer Fig. 2.28. The spark-plug consists of a metal shell having two electrodes which are insulated from each other with an air gap. High tension current jumping from the supply electrode produces the necessary spark. Plugs are sometimes identified by the heat range or the relative temperature obtained during operation. The correct type of plug with correct width of gap between the electrodes are important factors. The spark-plug gap can be easily checked by means of a feeler gauge and set as per manufacturer's specifications. It is most important that while adjusting the spark plug it is the outer earthed electrode *i.e.*, tip which is moved in or out gradually for proper setting of the gap. No bending force should be applied on the centre-electrode for adjusting the gap as this can cause crack and fracture of insulation and the plug may become absolutely useless.

Porcelain is commonly used as insulating material in spark-plugs, as it is cheap and easy to manufacture. Mica can also be used as insulating material for spark-plugs. Mica, however, cannot withstand high temperatures successfully.

- **Operating Heat Range :**
  - — A spark-plug heat range is a measure of the plug's ability to transfer heat from the central electrode and insulator nose to the cylinder-head and cooling system.
  - — When the heat absorbed by the plug's central electrode and insulator nose exceeds the capability of the plug to dissipate this heat in the same time, then the plug will *overheat* and the central electrode temperature will rise above its safe operating limit of about 900 to 950°C. *Above the plug's upper working temperature-limit, the central electrode will glow and ignite the air-fuel mixture before the timed spark actually occurs.* This condition is *known as* **auto-ignition** as it *automatically starts the combustion process independently of the controlled ignition spark.* The danger of this occuring is in the fact that it may take place relatively early in the compression stroke. Consequently, the pressure generated in the particular cylinder suffering from auto-ignition will oppose the upward movement of the piston. Excessive mechanical stresses will be produced in the reciprocating and rotating components and an abnormal rise in the cylinder temperature would, if allowed to continue, damage the engine.
  - — If the plug's ability to transfer heat away from the central electrode and insulator tip *exceeds* that of the input heat from combustion, over the same time span, then the plug's central electrode and insulator nose *would operate at such a low temperature as to permit the formation of carbon deposits around the central nose of the plug.* This *critical lower temperature region is usually between 350°C and 400°C* and, at temperatures below this, carbon or oil deposits will *foul the insulation*, creating conducting shunts to the inside of the metal casing of the plug. Consequently, if *deposits are permitted to form, a proportion of the ignition spark energy will bypass the plug gap so that there will be insufficient energy left to ionize the electrode with the result that misfiring will result.* Establishing a heat balance between the plug's input and output heat flow, so that the *plug's temperature remains just in excess of 400°C, provides a self cleaning action on both the surfaces of the electrodes and insulator.*
  - — A good spark-plug design tries to match the heat flowing from the plug to the heat flowing into it, caused by combustion under all working conditions, so that the plug operates below the upper temperature limit at full load, but never drops below the lower limit when idling or running under light-load conditions.

- **Firing Voltage :**

  A certain *minimum voltage* is necessary to make the spark jump the electrode air gap, the actual magnitude of the voltage required will depend upon the following *factors* :

  (*i*) Compression pressure (*ii*) Mixture strength
  (*iii*) Electrode gap (*iv*) Electrode tip temperature.

- **Tightness of Spark-plug :**
  - — The seat joint tightness is essential for good heat dissipation.
  - — Spark-plugs should not be over tightened otherwise the plug metal casing may become distorted, causing the central electrode insulator to break its seal and become loose. *Combustion gas may then escape through the plug with the result that it overheats.*

— An under-tightened plug may work itself loose and cause combustion gas to escape between the plug and cylinder-head plug hole threads to the atmosphere, again this will *result in overheating and rapid deterioration of the electrode tips.*

It is best to torque the plug to a definite degree of tightness.

Simple carburettor

The function of a carburettor is to atomise and metre the liquid fuel and mix it with the air as it enters the induction system of the engine, maintaining under all conditions of operation fuel-air proportions appropriate to those conditions.

All modern carburettors are based upon Bernoulli's theorem,

$$C^2 = 2gh$$

where C is the velocity in metres/sec, g is the acceleration due to gravity in metre/sec^2 and h is the head causing the flow expressed in metres of height of a column of the fluid.

The equation of mass rate of flow is given by,

$$m = \rho A \sqrt{2gh}$$

where ρ is the density of the fluid and A is the cross-sectional area of fluid stream.

In Fig. 2.29 is shown simple carburettor. L is the float chamber for the storage of fuel. The fuel supplied under gravity action or by fuel pump enters the float chamber through the filter F.

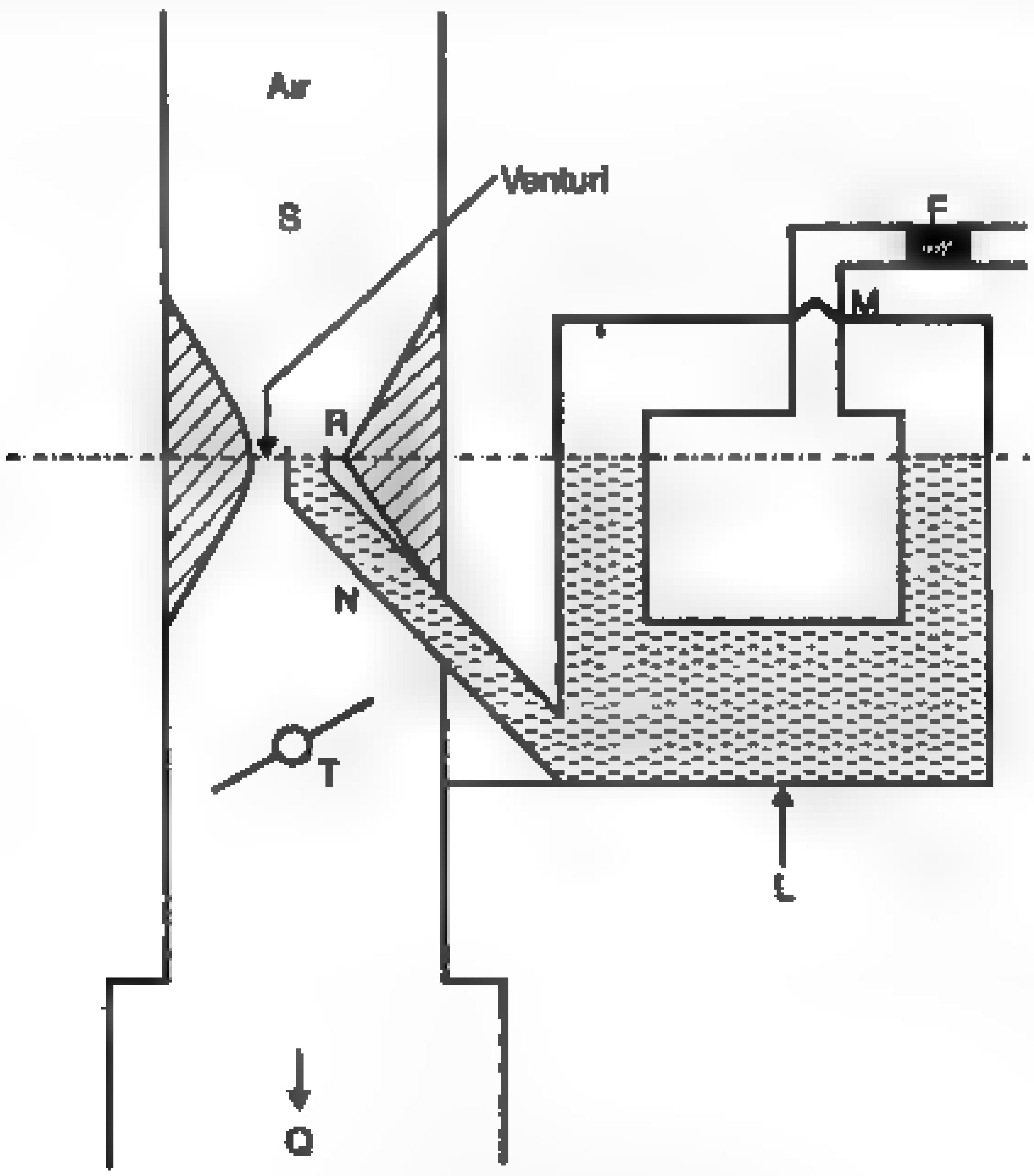


Fig. 2.29. Simple carburettor.

The arrangement is such that when the oil reaches a particular level the float valve *M* blocks the inlet passage and thus cuts off the fuel oil supply. On the fall of oil level, the float descends down, consequently intake passage opens and again the chamber is filled with oil Then the float and the float valve maintains a constant fuel oil level in the float chamber. *N* is the jet from which the fuel is sprayed into the air stream as it enters the carburettor at the inlet *S* and passes through the throat or venturi *R*. The fuel level is slightly below the outlet of the jet when the carburettor is inoperative.

As the piston moves down in the engine cylinder, suction is produced in the cylinder as well as in the induction manifold *Q* as a result of which air flows through the Carburettor. The velocity of air increases as it passes through the construction at the venturi *R* and pressure decreases due to conversion of a portion of pressure head into kinetic energy. Due to decreased pressure at the venturi and hence by virtue of difference in pressure (between the float chamber and the venturi) the jet issues fuel oil into air stream. Since the jet has a very fine bore, the oil issuing from the jet is in the form of fine spray ; it vapourises quickly and mixes with the air. This air fuel mixture enters the engine cylinder ; its quantity being controlled by varying the position of the throttle valve *T*.

Limitations :

(*i*) Although theoretically the air fuel ratio supplied by a simple (single jet) carburettor should remain constant as the throttle goes on opening, actually it provides increasingly richer mixture as the throttle is opened. This is because of the reason that the density of air tends to decrease as the rate of flow increases.

(*ii*) During idling, however, the nearly closed throttle causes a reduction in the mass of air flowing through the venturi. At such low rates of air flow, the pressure difference between the float chamber and the fuel discharge nozzle becomes very small. It is sufficient to cause fuel to flow through the jet.

(*iii*) Carburettor does not have arrangement for providing rich mixture during starting and warm up.

In order to *correct for faults* :

(*i*) number of compensating devices are used for (*ii*) an idling jet is used which helps in running the engine during idling. For (*iii*) choke arrangement is used.

Fuel pump (for carburettor-petrol engine).

Refer Fig. 2.30. This type of pump is used in petrol engine for supply of fuel to the carburettor. Due to rotation of the crankshaft the cam pushes the lever in the upward direction. One end of the lever is hinged while the other end pulls the diaphragm rod with the *diaphragm*. So the diaphragm comes in the downward direction against the compression of the spring and thus a vacuum is produced in the pump chamber. This causes the fuel to enter into the pump chamber from the *glass bowl* through the *strainer* and the inlet valve, the impurities of the fuel ; if there is any, deposit at the bottom of the glass bowl. On the return stroke the spring pushes the diaphragm in the upward direction forcing the fuel from the pump chamber into the carburettor through the *outlet valve*.

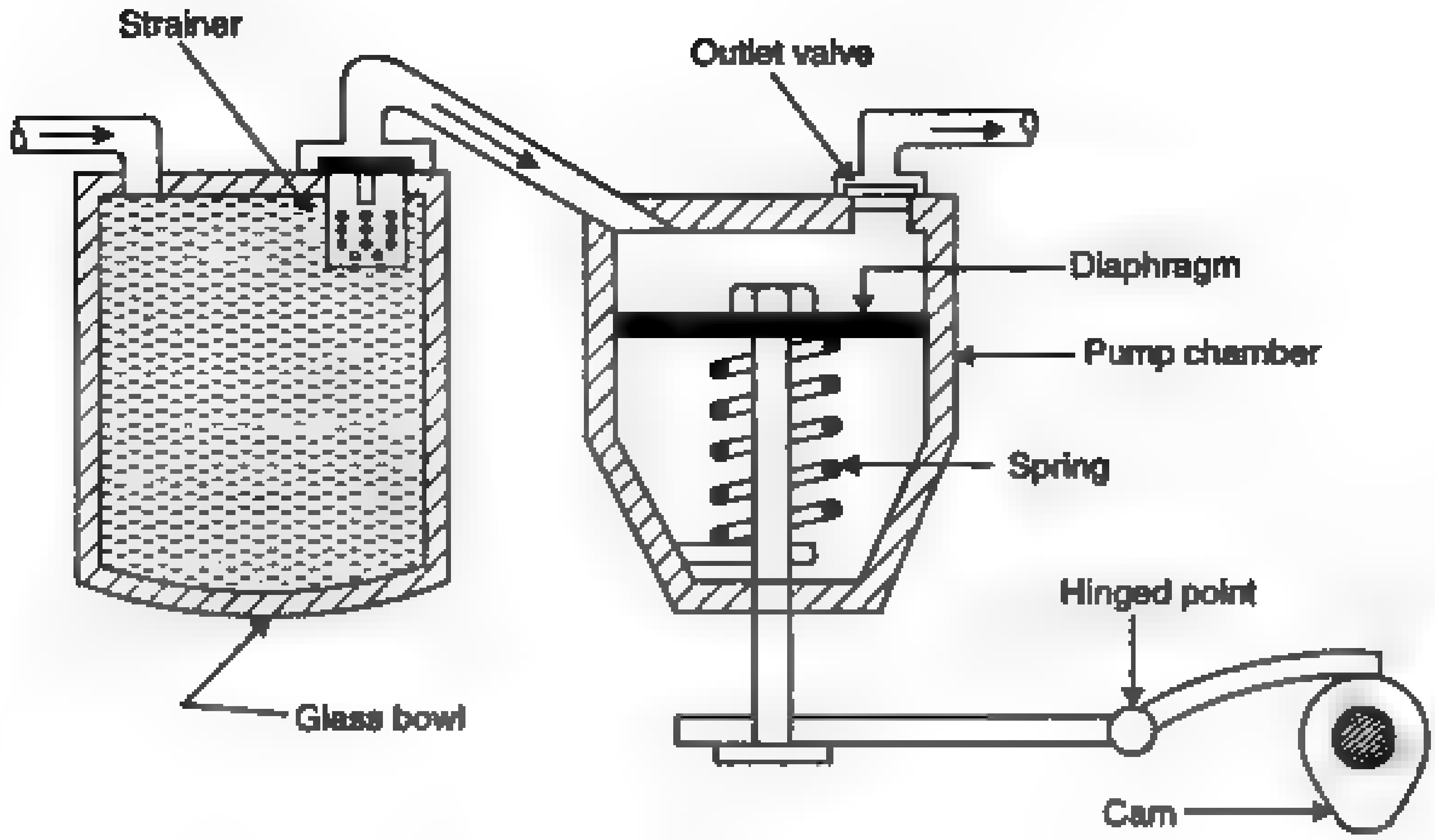


Fig. 2.30. Fuel pump for carburettor.

Parts for Diesel engine only :

FUEL PUMP

Refer Fig. 2.31. *L* is the plunger which is driven by a cam and tappet mechanism at the bottom (not shown in the figure) *B* is the barrel in which the plunger reciprocates. There is the rectangular vertical groove in the plunger which extends from top to another helical groove. *V* is the delivery valve which lifts off its seat under the liquid fuel pressure and against the spring force (*S*). The fuel pump is connected to fuel atomiser through the passage *P*, *SP* and *Y* are the spill and supply ports respectively. When the plunger is at its bottom stroke the ports *SP* and *Y* are uncovered (as shown in the Fig. 2.31) and oil from low pressure pump (not shown) after being filtered is forced into the barrel. When the plunger moves up due to cam and tappet mechanism, a stage reaches when both the ports *SP* and *Y* are closed and with the further upward movement of the plunger the fuel gets compressed. The high pressure thus developed lifts the delivery valve off its seat and fuel flows to atomiser through the passage *P*. With further rise of the plunger, at a certain moment, the port *SP* is connected to the fuel in the upper part of the plunger through the rectangular vertical groove by the helical groove ; as a result of which a sudden drop in pressure occurs and the delivery valve falls back and occupies its seat against the spring force. The plunger is rotated by the rack *R* which is moved in or out by the governor. *By changing the angular position of the helical groove (by rotating the plunger) of the plunger relative to the supply port, the length of stroke during which the oil is delivered can be varied and thereby quantity of fuel delivered to the engine is also varied accordingly.*

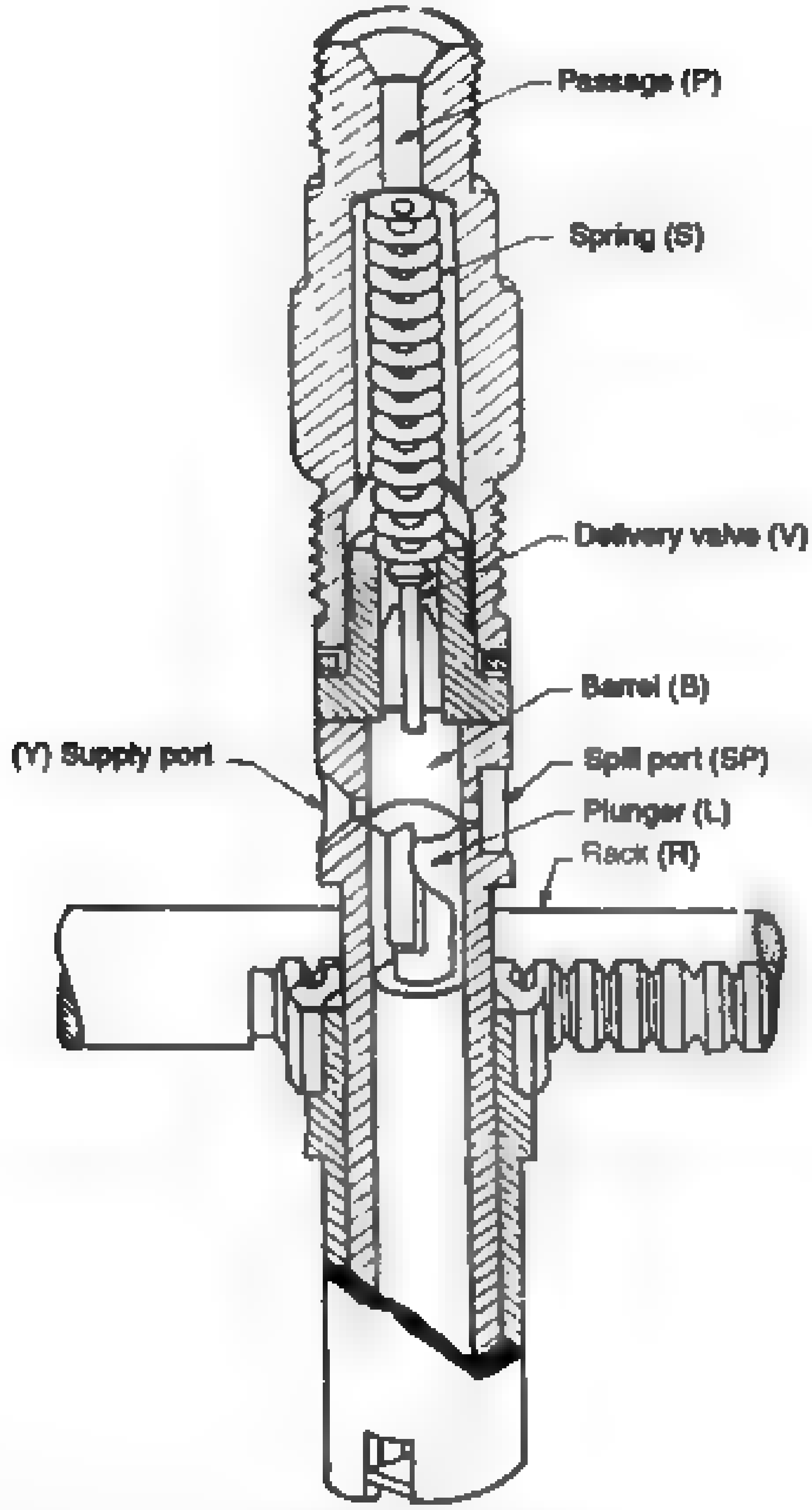


Fig. 2.31. Fuel pump.

Fuel atomiser or injector

Refer Fig. 2.32. It consists of a nozzle valve (*NV*) fitted in the nozzle body (*NB*). The nozzle valve is held on its seat by a spring '*S*' which exerts pressure through the spindle *E*. '*AS*' is the adjusting screw by which the nozzle valve lift can be adjusted. Usually the nozzle valve is set to lift at 135 to 170 bar pressure. *FP* is the feeling pin which indicates whether valve is working properly or not. The oil under pressure from the fuel pump enters the injector through the passages *B* and *C* and lifts the nozzle valve. The fuel travels down the nozzle *N* and injected into the engine cylinder in the form of fine sprays. When the pressure of the oil falls, the nozzle valve occupies its seat under the spring force and fuel supply is cut off. Any leakage of fuel accumulated above the valve is led to the fuel tank through the passage *A*. The leakage occurs when the nozzle valve is *worn out*.

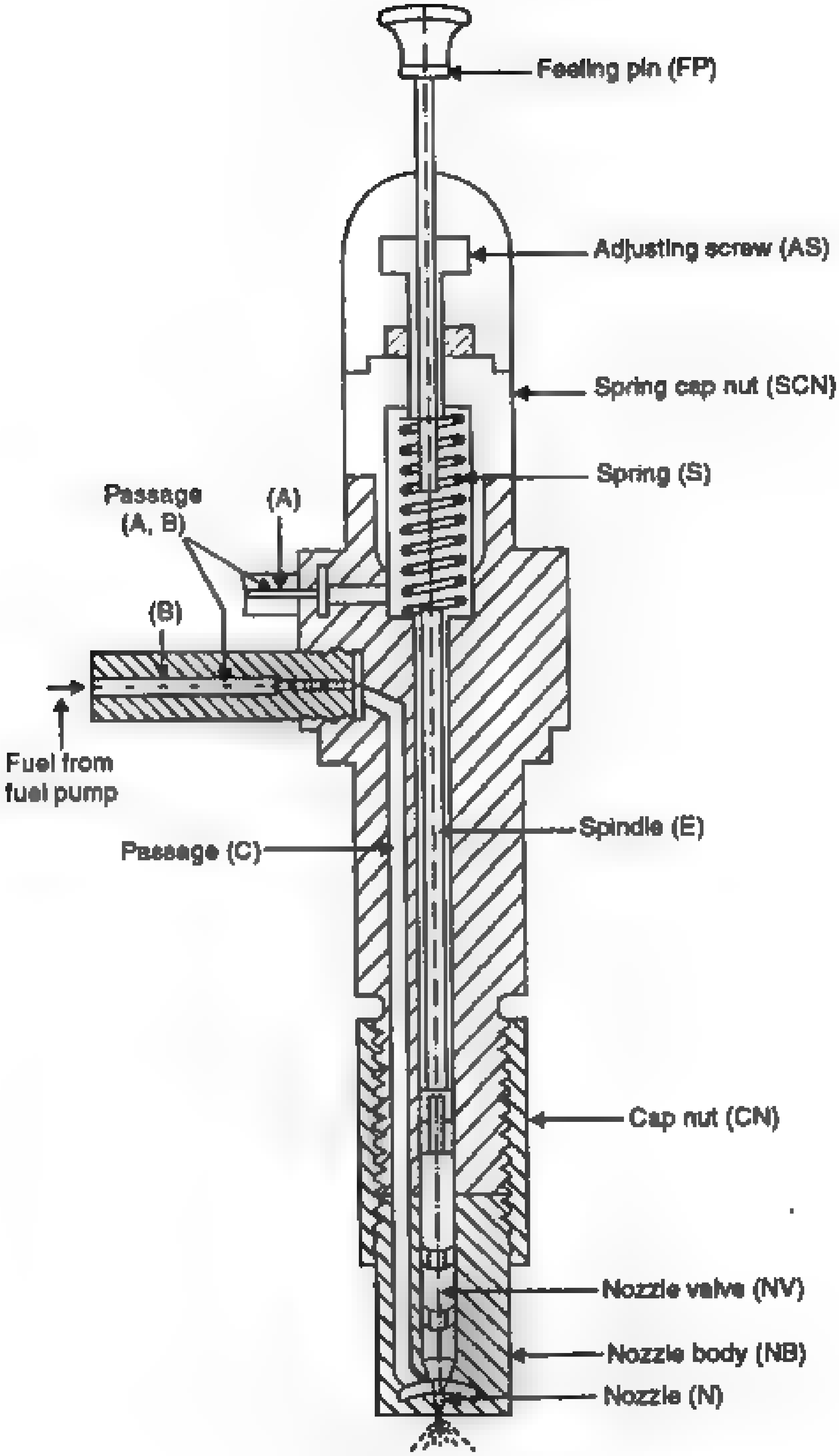


Fig. 2.32. Fuel atomiser or injector.

List of engine parts, materials, method of manufacture and functions :

	Name of the part	*Material*	*Function*	*Method of manufacture*
1.	*Cylinder*	Hard grade cast iron	Contains gas under pressure and guides the piston.	Casting
2.	*Cylinder head*	Cast iron or aluminium	Main function is to seal the working end of the cylinder and not to permit entry and exit of gases on overhead valve engines.	Casting, forging
3.	*Piston*	Cast iron or aluminium alloy	It acts as a face to receive gas pressure and transmits the thrust to the connecting rod.	Casting, forging
4.	*Piston rings*	Cast iron	Their main function is to provide a good sealing fit between the piston and cylinder.	Casting
5.	*Gudgeon pin*	Hardened steel	It supports and allows the connecting rod to swivel.	Forging
6.	*Connecting rod*	Alloy steel ; for small engines the material may be aluminium	It transmits the piston load to the crank, causing the latter to turn, thus converting the reciprocating motion of the piston into rotary motion of the crankshaft.	Forging
7.	*Crankshaft*	In general the crankshaft is made from a high tensile forging, but special cast irons are sometimes used to produce a light weight crank shaft that does not require a lot of machining.	It converts the reciprocating motion of the piston into the rotary motion.	Forging
8.	*Main bearings*	The typical bearing half is made of steel or bronze back to which a lining of relatively soft bearing material is applied.	The function of bearing is to [redacted] the parts to move easily.	Casting
9.	*Flywheel*	Steel or cast iron.	In engines it takes care of fluctuations of speed during thermodynamic cycle.	Casting
10.	*Inlet valve*	Silicon chrome steel with about 3% carbon.	Admits the air or mixture of air and fuel into engine cylinder.	Forging
11.	*Exhaust valve*	Austenitic steel	Discharges the product of combustion.	Forging

2.8. TERMS CONNECTED WITH I.C. ENGINES

Refer Fig. 2.33.

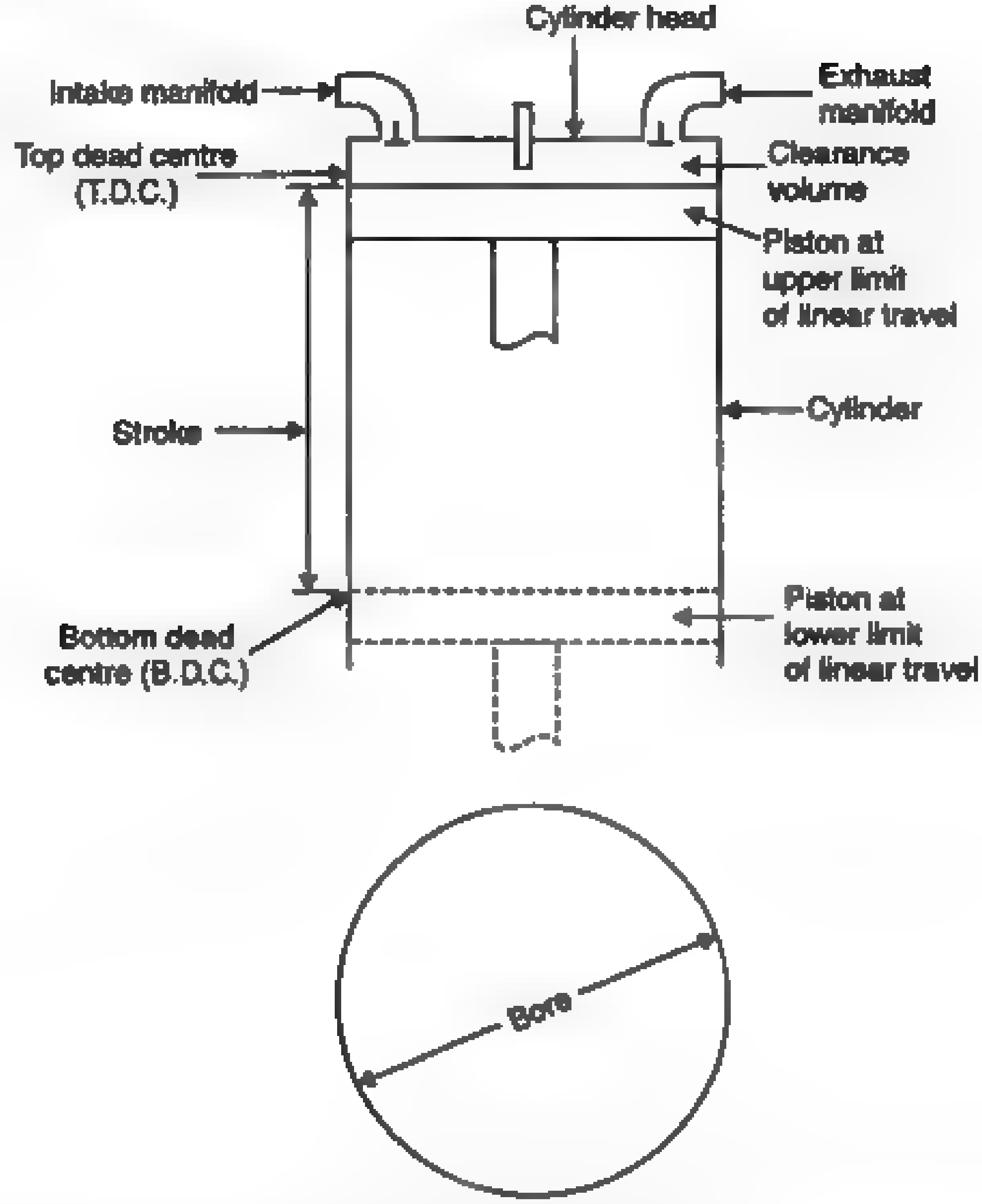


Fig. 2.33. Terms relating I.C. engines.

Bore. *The inside diameter of the cylinder is called "bore".*

Stroke. As the piston reciprocates inside the engine cylinder, it has got limiting upper and lower positions beyond which it cannot move and reversal of motion takes place at these limiting positions.

The linear distance along the cylinder axis between two limiting positions, is called "stroke".

Top Dead Centre (T.D.C.). *The top most position of the piston towards cover end side of the cylinder is called "top dead centre".* In case of horizontal engines, this is known as *inner dead centre.*

Bottom Dead Centre (B.D.C.). *The lowest position of the piston towards the crank end side of the cylinder is called "bottom dead centre".* In case of horizontal engines it is called *outer dead centre.*

Clearance volume. *The volume contained in the cylinder above the top of the piston, when the piston is at top dead centre, is called the "clearance volume".*

- Bore sizes of engines range from 0.5 m down to 0.5 cm. The ratio of bore of stroke D/L, for small engines is usually from 0.8 to 1.2.

— An engine with $L = D$ is often called a **square** *engine* ;
— If $L > D$ the engine is **under square** ;
— If $L < D$ the engine is **over square.**

Very large engines are *always under square*, with *stroke lengths up to four times bore diameter*.

Swept volume. *The volume swept through by the piston in moving between top dead centre and bottom dead centre, is, called "swept volume or piston displacement"*. Thus, when piston is at bottom dead centre, total volume = swept volume + clearance volume.

- Typical values for engine displacement range from 0.1 cm^3 for small model airplanes to about 8 litres for large automobiles to much large number for large ship engines. The displacement of a modern average automobile engine is about two to three litres.
- For a given displacement volume, a longer stroke allows for a smaller bore (under square), resulting in less surface area in the combustion chamber and correspondingly less heat loss. This increases thermal efficiency within the combustion chamber. However, the longer stroke results in higher piston speed and higher friction losses that reduce the output power which can be obtained off the crankshaft. If the stroke is shortened, the bore must be increased and the engine will be over square. This decreases friction losses but increases heat transfer losses. *Most modern automobile engines are near square, with some slightly over square and some slightly under square.*

Compression ratio. *It is ratio of total cylinder volume to clearance volume.*

Refer Fig. 2.33. Compression ratio (r) is given by

$$r = \frac{V_s + V_c}{V_c}$$

where V_s = Swept volume, V_c = Clearance volume.

The compression ratio varies from 5 : 1 to 11 : 1 (average value 7 : 1 to 9 : 1) in *S.I. engines* and from 12 : 1 to 24 : 1 (average value 15 : 1 to 18 : 1) in *C.I. engines*.

- Modern spark ignition (S.I.) engines have compression ratios of 8 to 11, while compression ignition (C.I.) engines have compression ratios in the range 12 to 24. *Engines with superchargers or turbochargers usually have lower compression ratios than naturally aspirated engines.*
- Various attempts have been made to develop engines with a *variable compression ratio*. One such system uses a *split piston that expands due to changing hydraulic pressure caused by engine speed and load.* Some two-stroke cycle engines have been built which have a sleeve-type valve that changes the slot opening on the exhaust port. The piston where the exhaust port is fully closed can be adjusted by several degrees of engine rotation. This changes the *effective compression ratio* of the engine.

Piston speed. *The average speed of the piston is called "piston speed".*

Piston speed = 2 LN

where L = Length of the stroke, and

N = Speed of the engine in r.p.m.

- Average engine speed for all engines will normally be in the range of 5 to 15 m/s with large diesel engines on the low end and high performance automobile engines on the high end. There are following *two reasons* why engines operate in this range :
  — *First*, this is about the *safe limit which can be tolerated by material strength of the engine components.*
  — The *second* reason why maximum average piston speed is limited is *because of the gas flow into and out of cylinders.* Piston speed determines the instantaneous flow

rate of air-fuel into the cylinder during intake and exhaust flow out of the cylinder during the exhaust stroke. *Higher piston speeds would require larger valves to allow for higher flow rates.* In most engines, valves are at a maximum size with no room for enlargement.

Some Other Terms :

Direct Injection (D.I.). *Fuel injection into the main combustion chamber of an engine.* Engines have either one main combustion chamber (open chamber) or a divided combustion chamber made up of a main chamber and a smaller connected secondary chamber.

Indirect Injection (I.D.I.). *Fuel injection into the secondary chamber of an engine with a divided combustion chamber.*

Smart Engine. *Engine with computer controls that regulate operating characteristics* such as *air-fuel ratio, ignition timing, valve timing, exhaust control, intake tuning* etc.

Engine Management System (E.M.S.). Computer and electronics used to control smart engines.

Wide Open Throttle (W.O.T.). Engine operated with *throttle valve fully open when maximum power and/or speed is desired.*

Ignition Delay (I.D.). It is the *time interval between ignition initiation and the actual start of combustion.*

Air-Fuel Ratio (A/F). It is the *ratio of the air to mass of fuel input into engine.*

Fuel-Air Ratio (F/A). It is the *ratio of fuel to mass of air input into engine.*

2.9. WORKING CYCLES

An internal combustion engine can work on any one of the following cycles :

(*a*) Constant volume or Otto cycle

(*b*) Constant pressure or Diesel cycle

(*c*) Dual combustion cycle.

These may be either *four stroke cycle or two stroke cycle engines.*

(*a*) **Constant volume or Otto cycle.** The cycle is so called because heat is supplied at constant volume. Petrol, gas, light oil engines work on this cycle. In the case of a petrol engine the proper mixing of petrol and air takes place in the carburettor which is situated outside the engine cylinder. The proportionate mixture is drawn into the cylinder during the suction stroke. In a gas engine also, air and gas is mixed outside the engine cylinder and this mixture enters the cylinder during the suction stroke. In light oil engines the fuel is converted to vapours by a vapouriser which receives heat from the exhaust gases of the engine and their mixture flows towards engine cylinder during suction stroke.

(*b*) **Constant pressure or diesel cycle.** In this cycle only air is drawn in the engine cylinder during the suction stroke, this air gets compressed during the compression stroke and its pressure and temperature increase by a considerable amount. Just before the end of the stroke a metered quantity of fuel under pressure adequately more than that developed in the engine cylinder is injected in the form of fine sprays by means of a fuel injector. Due to very high pressure and temperature of the air the fuel ignites and hot gases thus produced throw the piston downwards and work is obtained. *Heavy oil engines make use of this cycle.*

(*c*) **Dual combustion cycle.** This cycle is also called *semi-diesel cycle.* It is so named because heat is added *partly at constant volume and partly at constant pressure.* In this cycle only air is drawn in the engine cylinder during suction stroke. The air is then compressed in hot combustion chamber at the end of the cylinder during the compression stroke to a pressure of about 26 bar. The heat of compressed air together with heat of combustion chamber ignites the fuel. The fuel is injected into the cylinder just before the end of compression stroke where it ignites

immediately. The fuel injection is continued until the point of cut off is reached. The burning of fuel at first takes place at constant volume and continues to burn at constant pressure during the first part of expansion or working stroke. The field of application of this cycle is heavy oil engines.

2.10. INDICATOR DIAGRAM

An indicator diagram is a graph between pressure and volume ; the former being taken on vertical axis and the latter on the horizontal axis. This is obtained by an instrument known as *indicator*. The indicator diagrams are of two types : (*a*) Theoretical or hypothetical, (*b*) Actual. The theoretical or hypothetical indicator diagram is always longer in size as compared to the actual one, since in the former losses are neglected. The ratio of the area of the actual indicator diagram to the theoretical one is called *diagram factor*.

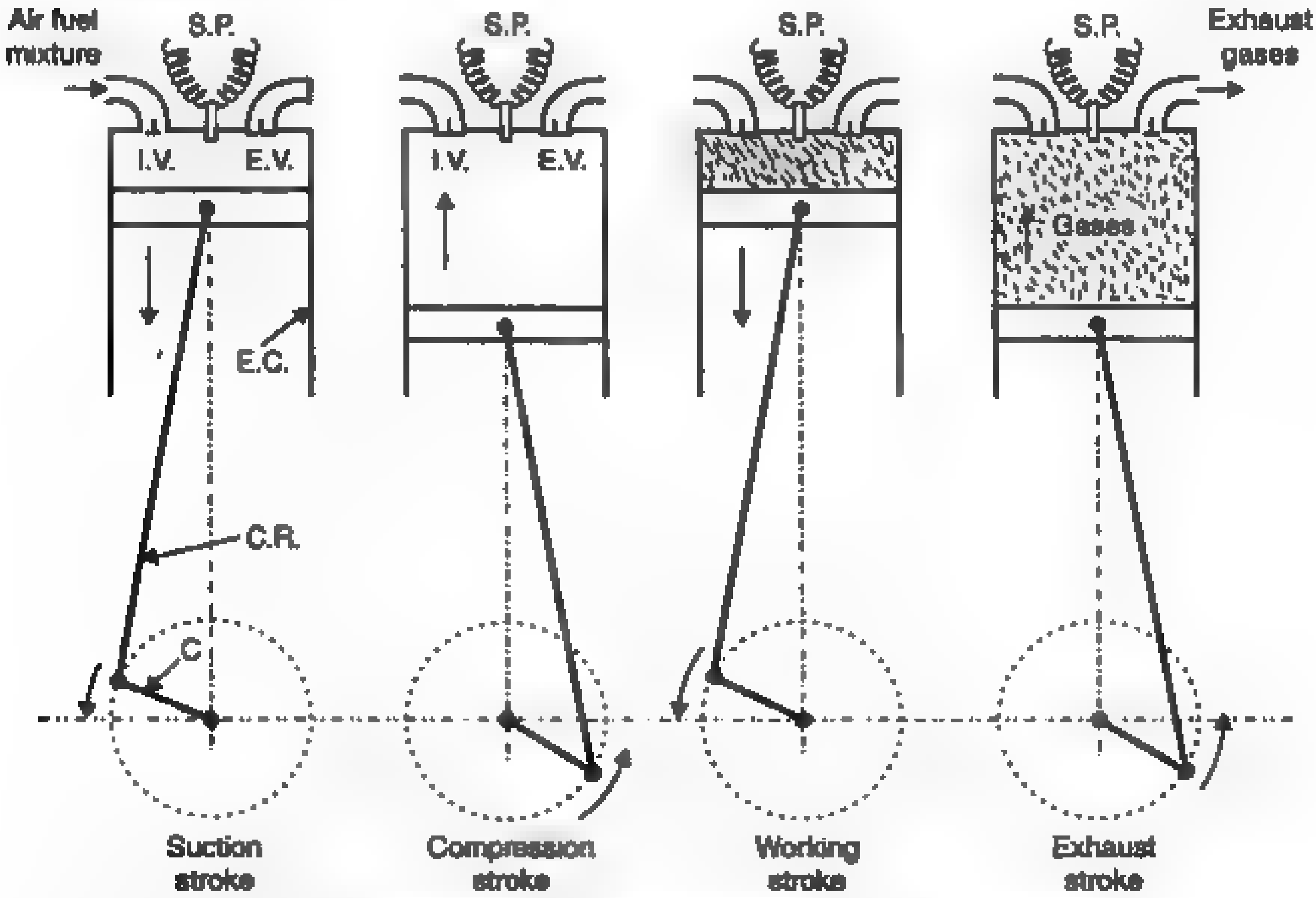
2.11. FOUR STROKE CYCLE ENGINES

Here follows the description of the four stroke otto and diesel-cycle engines.

Otto engines. The Otto four stroke-cycle refers to its use in petrol engines, gas engines, light oil engines in which the mixture of air and fuel are drawn in the engine cylinder. Since ignition in these engines is due to a spark, therefore they are also called *spark ignition engines*.

The various strokes of a four stroke (Otto) cycle engine are detailed below.

Refer Fig. 2.34.



I.V. = Inlet valve, E.V. = Exhaust valve, E.C. = Engine cylinder, C.R. = Connecting rod
C = Crank, S.P. = Spark plug.

Fig. 2.34. Four stroke Otto cycle engine.

1. **Suction stroke.** During this stroke (also known as induction stroke) the piston moves from top dead centre (T.D.C.) to bottom dead centre (B.D.C.) ; the inlet valve opens and proportionate fuel air mixture is sucked in the engine cylinder. This operation is represented by the line 5—1 (Fig. 2.32). The exhaust valve remains closed throughout the stroke.

2. **Compression stroke.** In this stroke, the piston moves (1—2) towards (T.D.C.) and compresses the enclosed fuel air mixture drawn in the engine cylinder during suction. The pressure of the mixture rises in the cylinder to a value of about 8 bar. Just before the end of this stroke the operating-plug initiates a spark which ignites the mixture and combustion takes place at constant volume (line 2—3) (Fig. 2.35). Both the inlet and exhaust valves remain closed during the stroke.

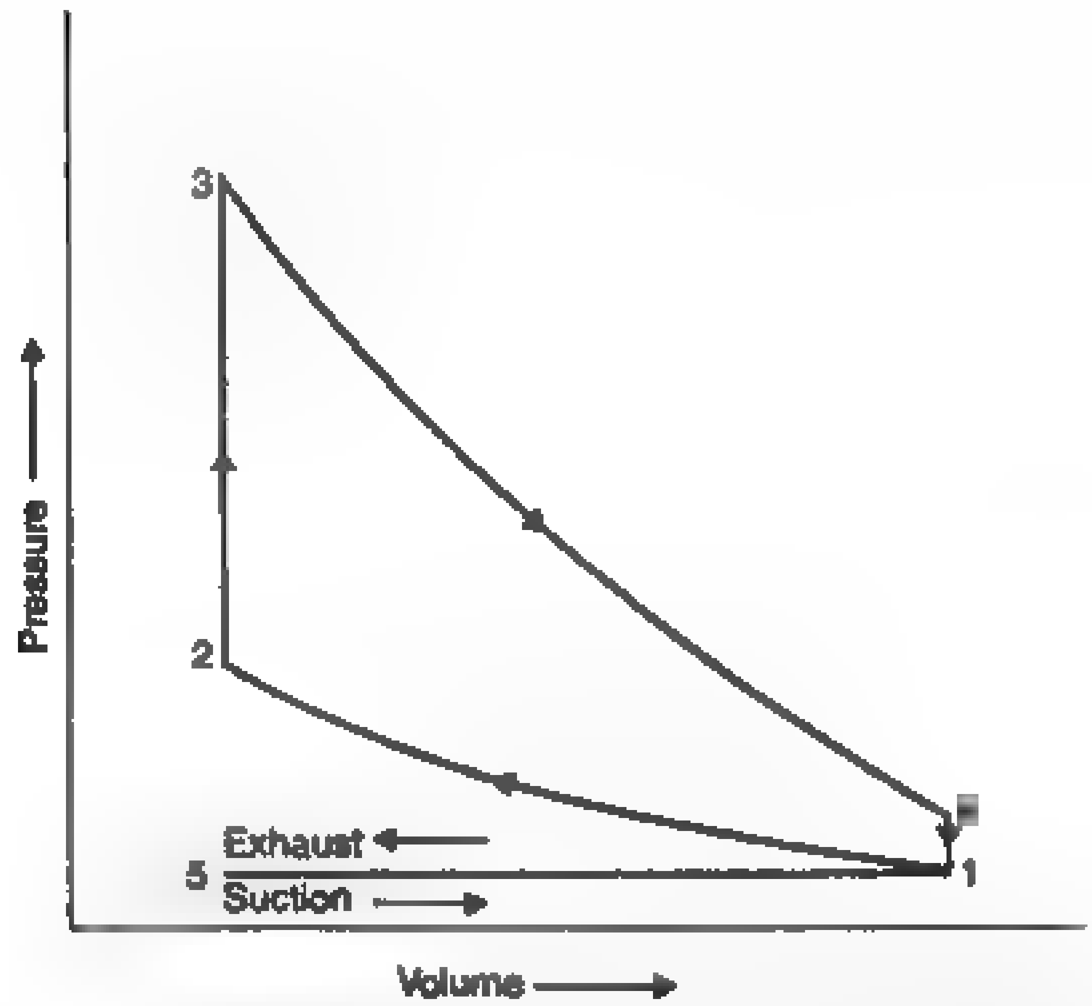


Fig. 2.35. Theoretical p-V diagram of a four stroke Otto cycle engine.

3. **Expansion or working stroke.** When the mixture is ignited by the spark plug the hot gases are produced which drive or throw the piston from T.D.C. to B.D.C. and thus the work is obtained in this stroke. It is during this stroke when we get work from the engine ; the other three strokes namely suction, compression and exhaust being idle. *The flywheel mounted on the engine shaft stores energy during this stroke and supplies it during the idle strokes.* The expansion of the gases is shown by 3-4. (Fig. 2.35). Both the valves remain closed during the start of this stroke but when the piston just reaches the B.D.C. the exhaust valve opens.

4. **Exhaust stroke.** This is the last stroke of the cycle. Here the gases from which the work has been collected become useless after the completion of the expansion stroke and are made to escape through exhaust valve to the atmosphere. This removal of gas is accomplished during this stroke. The piston moves from B.D.C. to T D.C. and the exhaust gases are driven out of the engine cylinder ; this is also called *scavenging*. This operation is represented by the line (1-5) (Fig. 2.35).

Fig. 2.36 shows the actual indicator diagram of four stroke Otto cycle engine. It may be noted that line 5-1 is below the atmospheric pressure line. This is due to the fact that owing to restricted area of the inlet passages the entering fuel air mixture cannot cope with the speed of the piston. The exhaust line 4-5 is slightly above the atmospheric pressure line. This is due to restricted exhaust passages which do not allow the exhaust gases to leave the engine-cylinder quickly.

The loop which has area 4-5-1 is called *negative loop* ; it gives the pumping loss due to admission of fuel air mixture and removal of exhaust gases. The area 1-2-3-4 is the total or gross work obtained from the piston and network can be obtained by subtracting area 451 from the area 1-2-3-4.

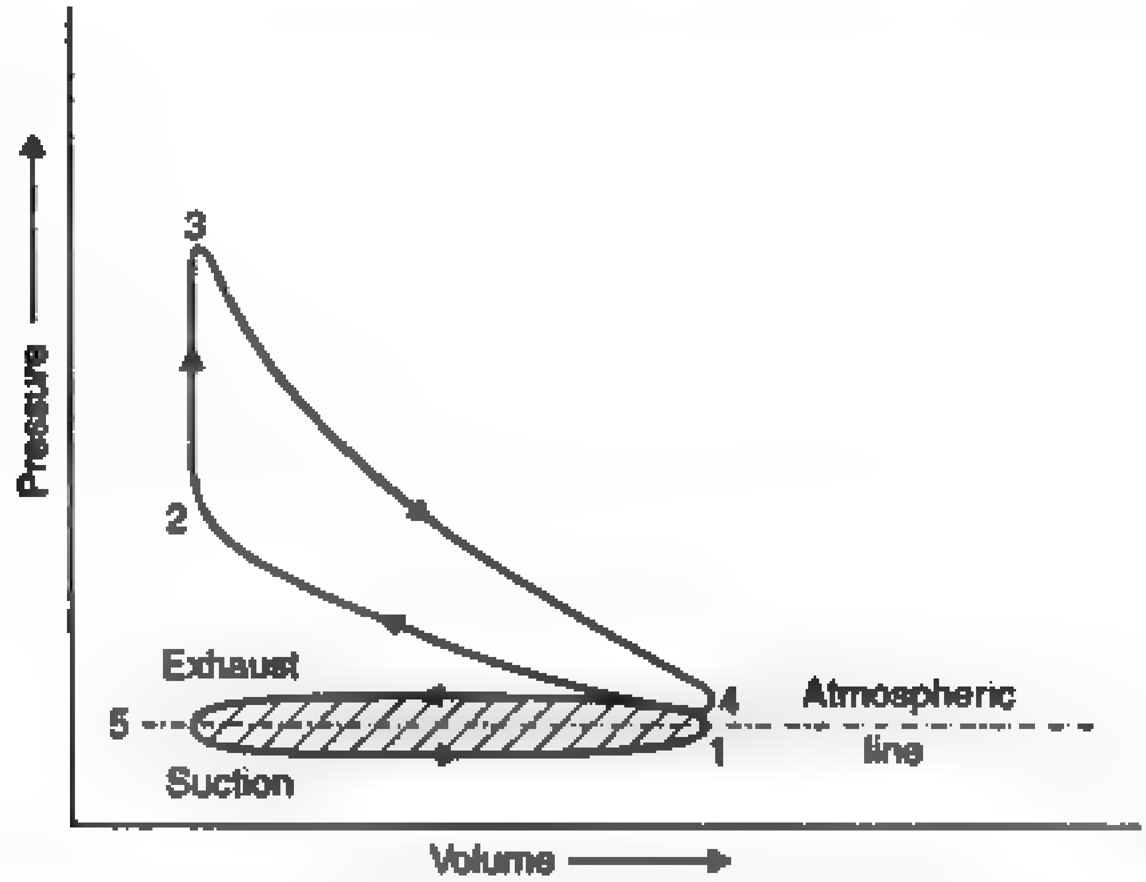


Fig. 2.36. Actual *p-V* diagram of a four stroke Otto cycle engine.

Diesel engines (four stroke cycle). As is the case of Otto four stroke ; this cycle too is completed in four strokes as follows. (Refer Fig. 2.37).

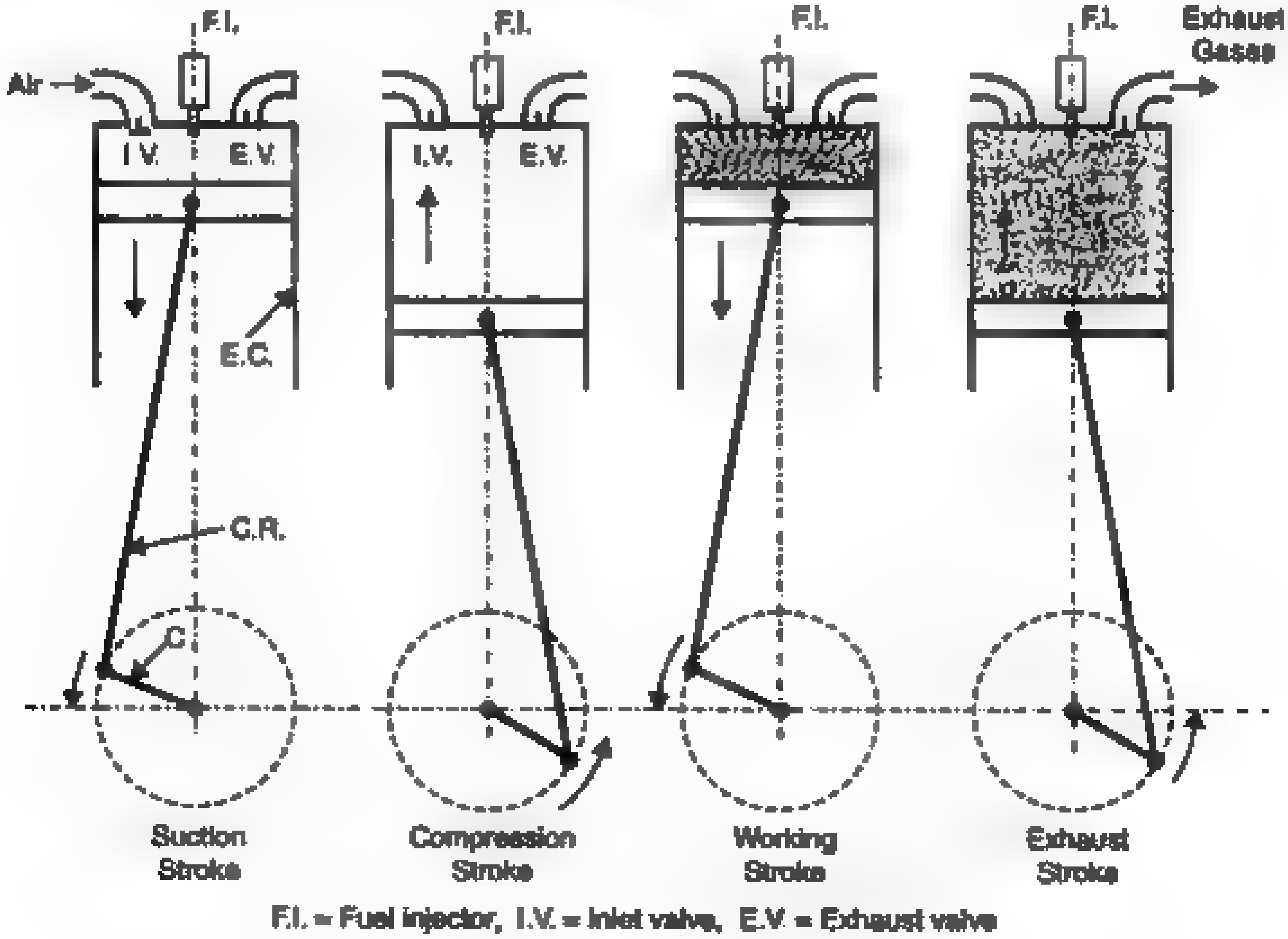


Fig. 2.37. Four stroke Diesel cycle engine.

1. **Suction stroke.** With the movement of the piston from T.D.C. to B.D.C. during this stroke, the inlet valve opens and the air at atmospheric pressure is drawn inside the engine cylinder ; the exhaust valve however remains closed. This operation is represented by the line 5-1 (Fig. 2.38).

2. **Compression stroke.** The air drawn at atmospheric pressure during the suction stroke is compressed to high pressure and temperature (to the value of 35 bar and 600°C respectively) as the piston moves from B.D.C. to T.D.C. This operation is represented by 1-2 (Fig. 2.38). Both the inlet and exhaust valves do not open during any part of this stroke.

3. **Expansion or working stroke.** As the piston starts moving from T.D.C. a metered quantity of fuel is injected into the hot compressed air in fine sprays by the fuel injector and it (fuel) starts burning at constant pressure shown by the line 2-3. At the point 3 fuel supply is cut off. The fuel is injected at the end of compression stroke but in actual practice the ignition of the fuel starts before the end of the compression stroke. The hot gases of the cylinder expand adiabatically to point 4, thus doing work on the piston. The expansion is shown by 3-4 (Fig. 2.38).

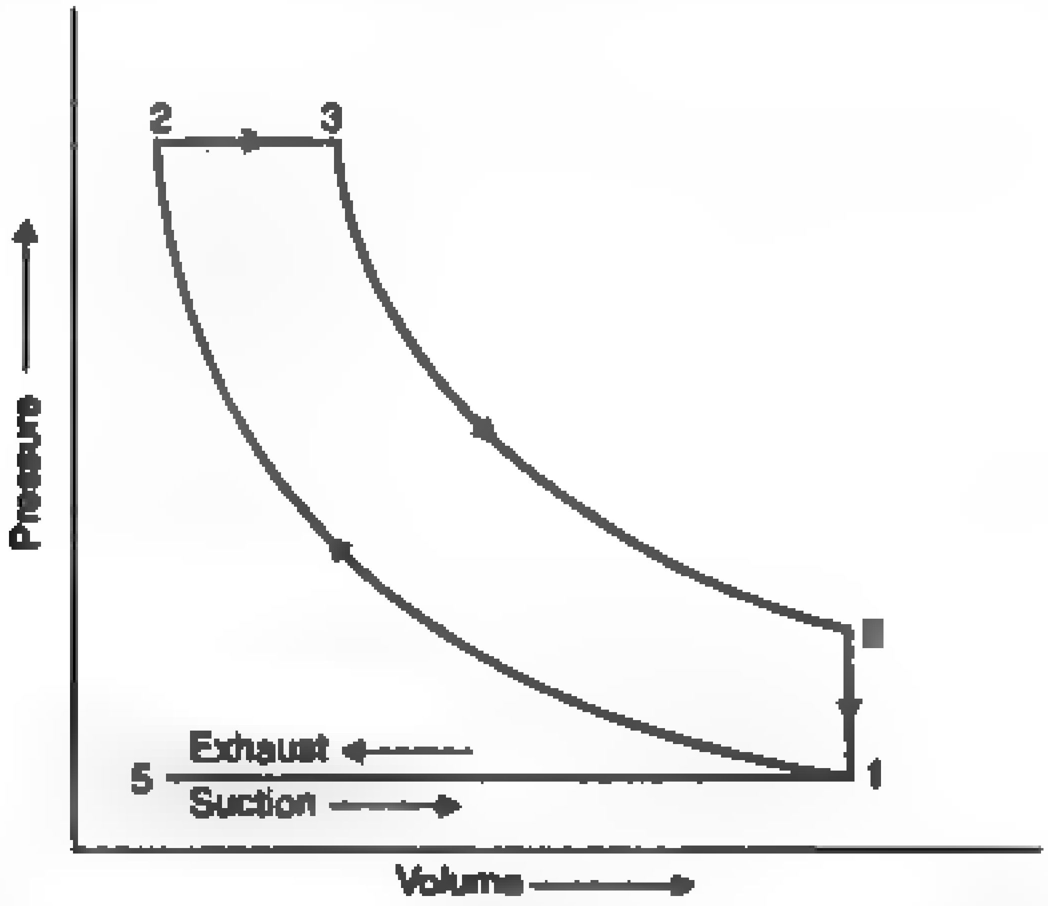


Fig. 2.38. Theoretical *p-V* diagram of a four stroke Diesel cycle.

4. **Exhaust stroke.** The piston moves from the B.D.C. to T.D.C. and the exhaust gases escape to the atmosphere through the exhaust valve. When the piston reaches the T.D.C. the exhaust valve closes and the cycle is completed. This stroke is represented by the line 1-5 (Fig. 2.38).

Fig. 2.39 shows the actual indicator diagram for a four-stroke Diesel cycle engine. It may be noted that line 5-1 is below the atmospheric pressure line. This is due to the fact that owing to the restricted area of the inlet passages the entering air can't cope with the speed of the piston. The exhaust line 4-5 is slightly above the atmospheric line. This is because of the restricted exhaust passages which do not allow the exhaust gases to leave the engine cylinder quickly.

The loop of area 4-5-1 is called negative loop ; it gives the pumping loss due to admission of air and removal of exhaust gases. The area 1-2-3-4 is the total or gross work obtained from the piston and net work can be obtained by subtracting area 4-5-1 from area 1-2-3-4.

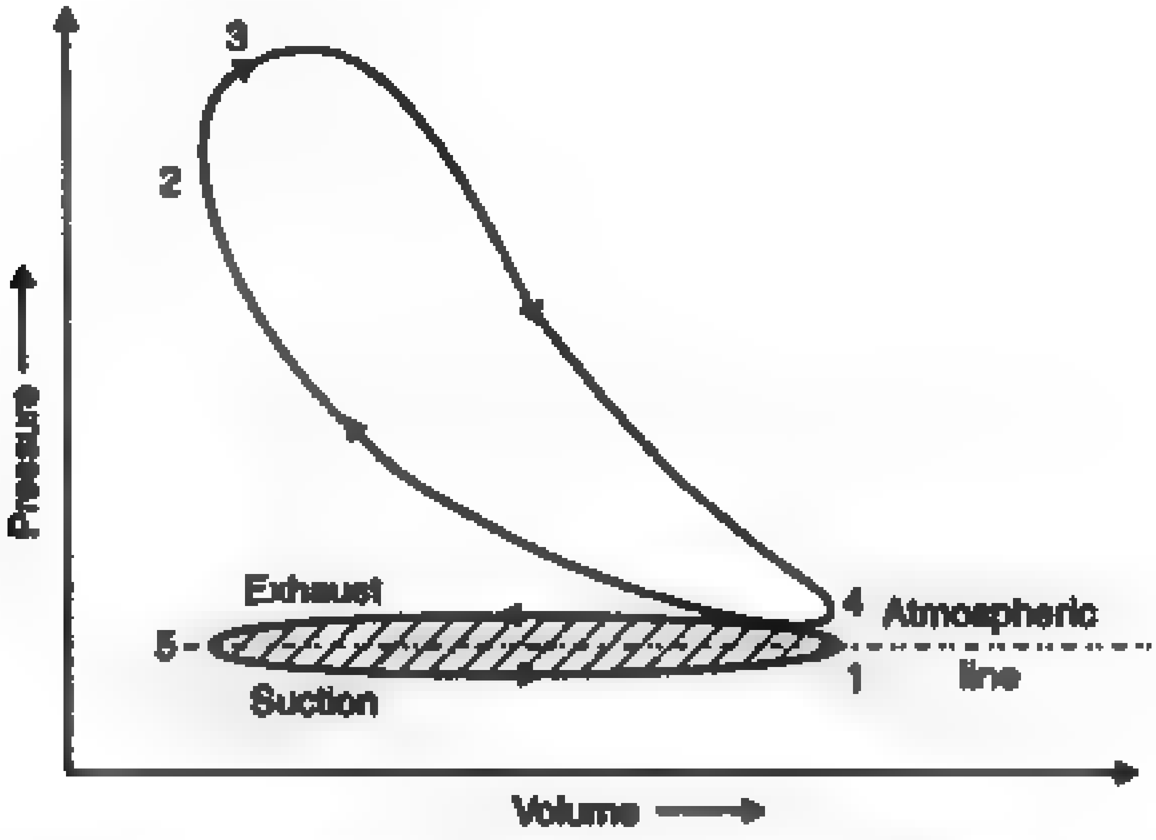


Fig. 2.39. Actual p-V diagram of four stroke Diesel cycle.

Valve Timing Diagrams (Otto and Diesel engines)

1. Otto engine. Fig. 2.40 shows a theoretical valve timing diagram for *four stroke *Otto*

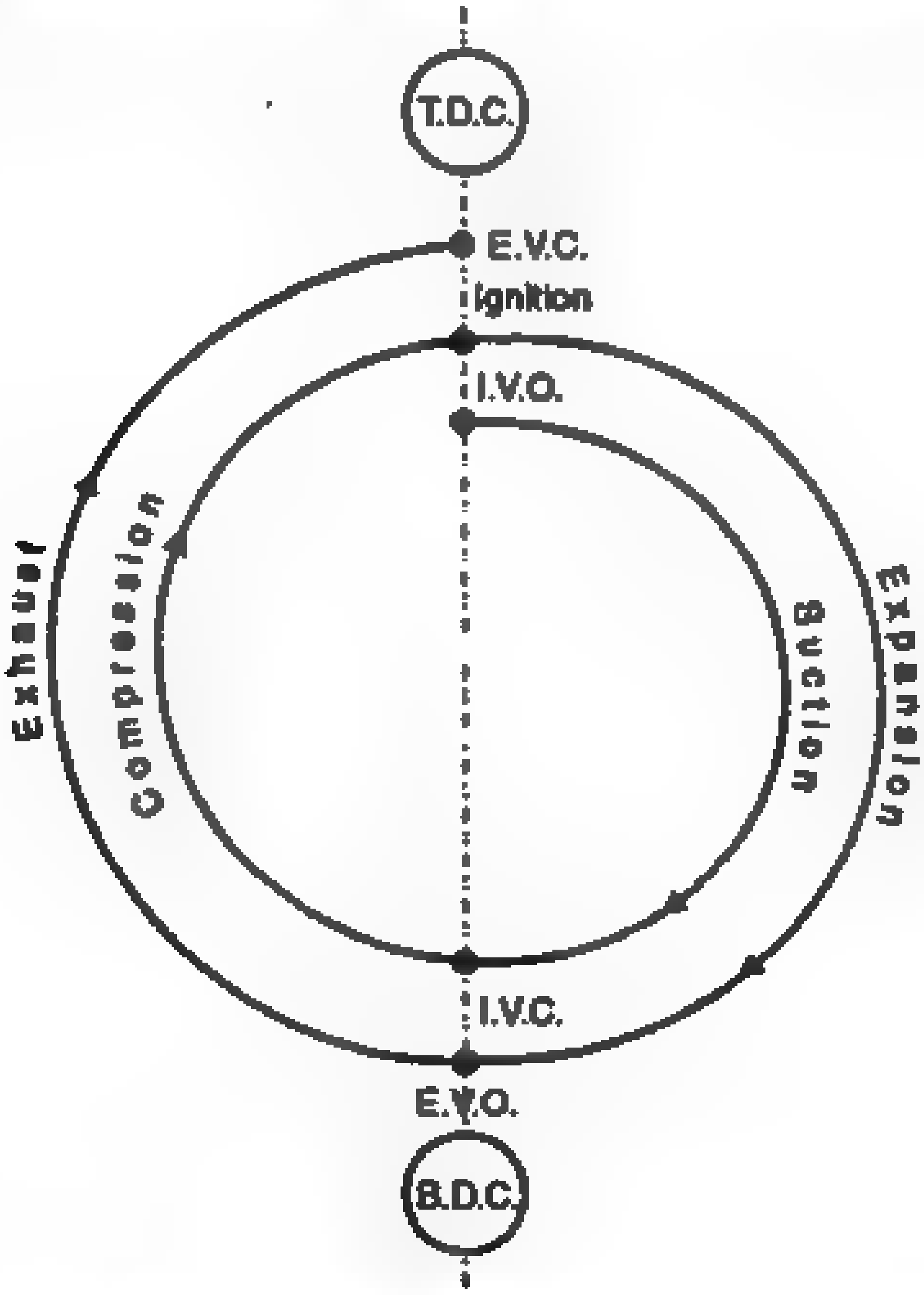


Fig. 2.40. Theoretical valve timing diagram (four stroke Otto cycle engine).

cycle" engines which is self-explanatory. In actual practice, it is difficult to open and close the valve instantaneously ; so as to get better performance of the engine the valve timings are modified. In Fig. 2.41 is shown an actual valve timing diagram. The inlet valve is opened 10° to 30° in advance of the T.D.C. position to enable the fresh charge to enter the cylinder and to help the burnt gases at the same time, to escape to the atmosphere. The suction of the mixture continues up to 30°-40° or even 60° after B.D.C. position. The inlet valve closes and the compression of the entrapped mixture starts.

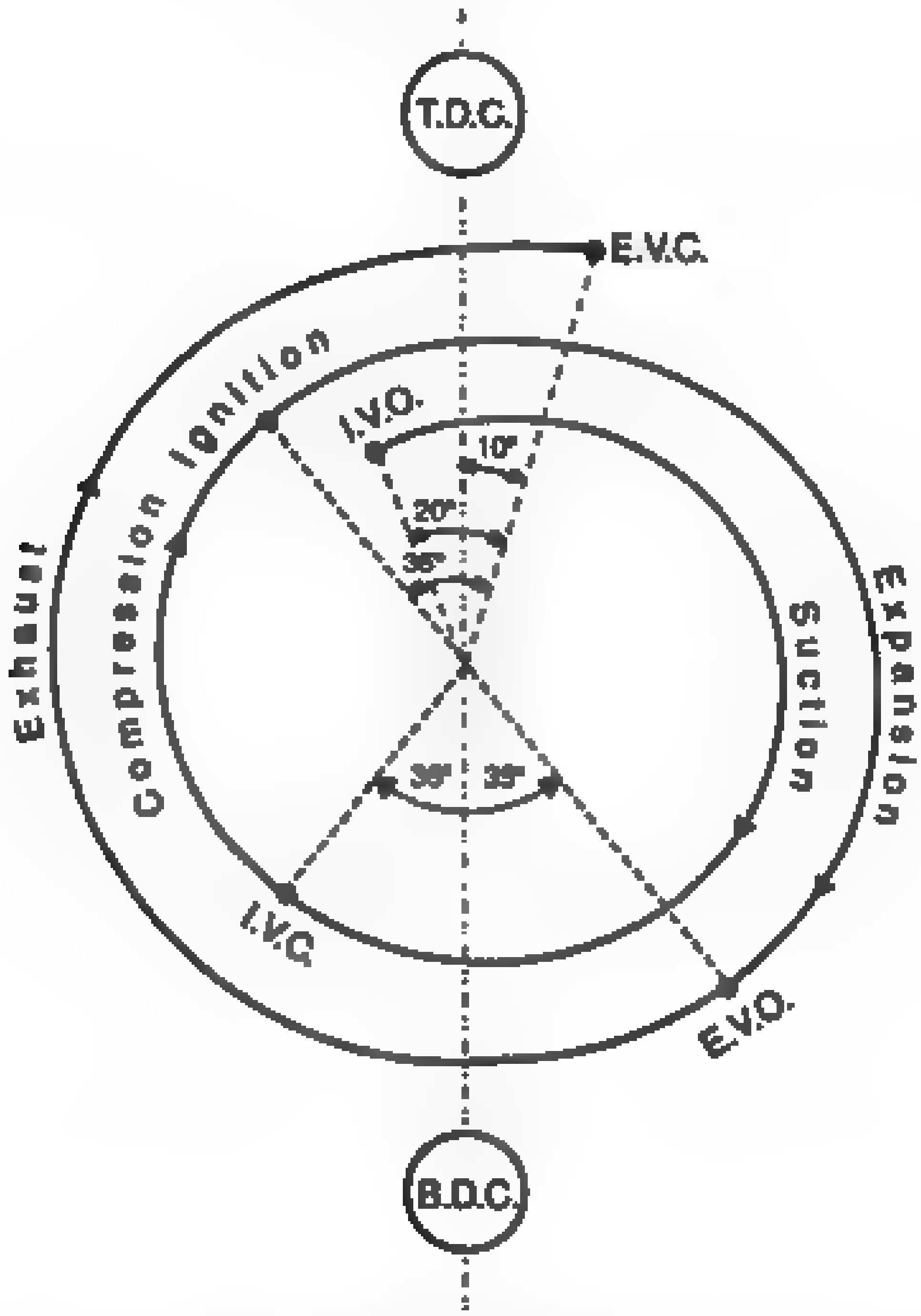


Fig. 2.41. Actual valve timing diagram (four Stroke Otto cycle engines).

The sparking plug produces a spark 30° to 40° before the T.D.C. position ; thus fuel gets more time to burn. The pressure becomes maximum nearly 10° past the T.D.C. position. The exhaust valve opens 30° to 60° before the B.D.C. position and the gases are driven out of the cylinder by piston during its upward movement. The exhaust valve closes when piston is nearly 10° past T.D.C. position.

2. **Diesel engines.** Fig. 2.42 shows the valve timing diagram of a *four stroke "Diesel cycle" engine* (theoretical valve timing diagram, is however the same as Fig. 2.40). Inlet valve opens 10° to 25° in advance of T.D.C. position and closes 25° to 50° after the B.D.C. position. Exhaust valve opens 30° to 50° in advance of B.D.C. position and closes 10° to 15° after the T.D.C. position. The fuel injection takes place 5° to 10° before T.D.C. position and continues up to 15° to 25° near T.D.C. position.

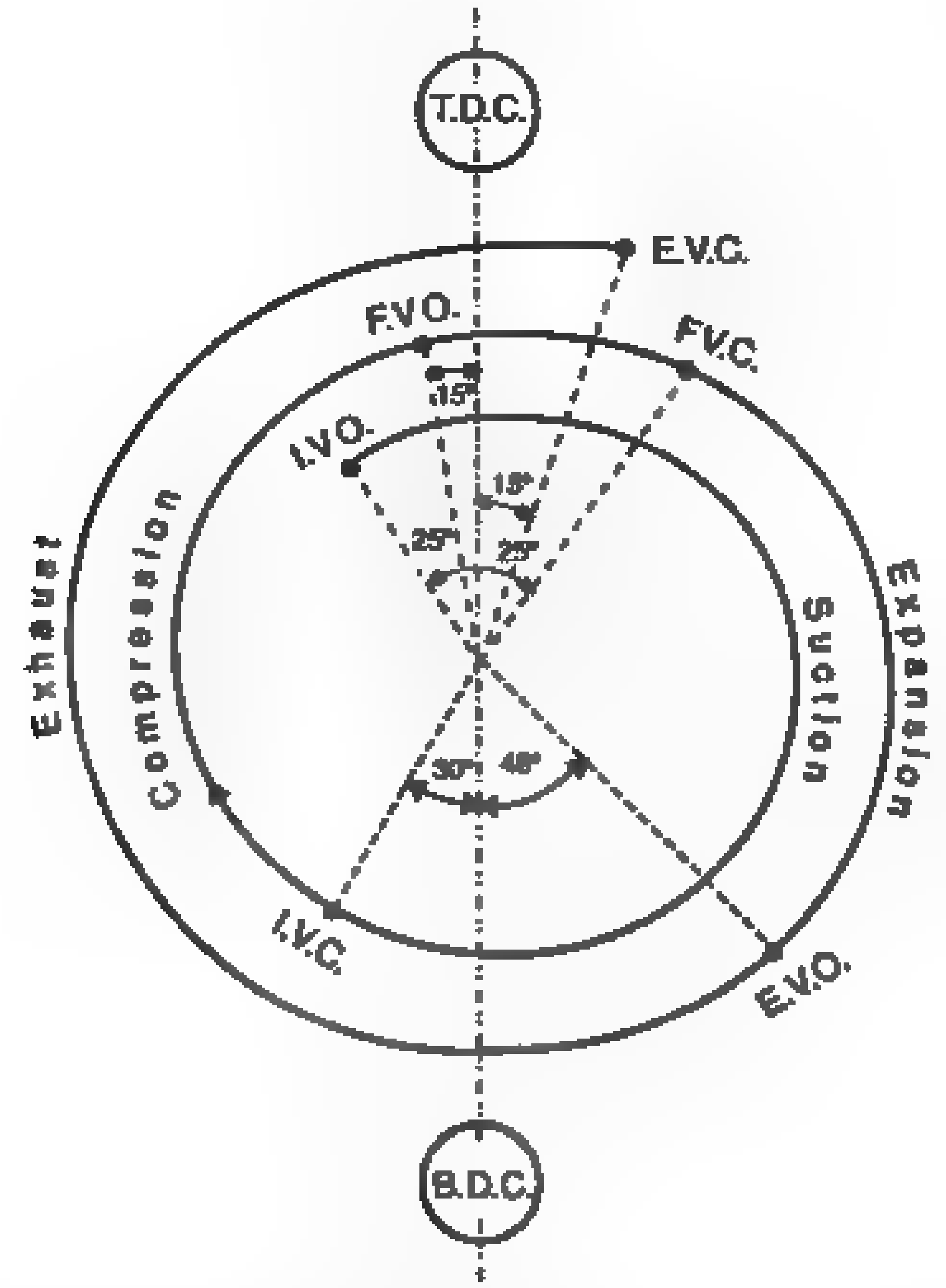


Fig. 2.42. Actual valve timing diagram (four stroke Diesel cycle engines).

[illegible] TWO STROKE CYCLE ENGINES

In 1878, Dugald-clerk, a British engineer introduced a cycle which could be completed in two *strokes of piston rather than four strokes* as is the case with the four stroke cycle engines. The engines using this cycle were called two stroke cycle engines. In this engine suction and exhaust strokes are eliminated. Here *instead of valves, ports are used. The exhaust gases are driven out from engine cylinder by the fresh charge of fuel entering the cylinder nearly at the end of the working stroke.*

Fig. 2.43 shows a two stroke petrol engine (used in scooters, motor cycles etc.). The cylinder L is connected to a closed crank chamber C.C. During the upward stroke of the piston M, the gases in L are compressed and at the same time fresh air and fuel (petrol) mixture enters the crank chamber through the valve V. When the piston moves downwards, V closes and the mixture in the crank chamber is compressed. Refer Fig. 2.43 (*i*), the piston is moving upwards and is compressing an explosive change which has previously been supplied to L. Ignition takes place at the end of the stroke. The piston then travels downwards due to expansion of the gases (Fig. 2.43 (*ii*)) and near the end of this stroke the piston uncovers the exhaust port (E.P.) and the burnt exhaust gases escape through this port (Fig. 2.43 (*iii*)). The transfer port (T.P.) then is uncovered immediately, and the compressed charge from the crank chamber flows into the cylinder and is deflected upwards by the hump provided on the head of the piston. It may be noted that the incoming air petrol

mixture helps the removal of gases from the engine-cylinder ; if, in case these exhaust gases do not leave the cylinder, the fresh charge gets diluted and efficiency of the engine will decrease. The piston then again starts moving from B.D.C. to T.D.C. and the charge gets compressed when E.P. (exhaust port) and T.P. are covered by the piston ; thus the cycle is repeated.

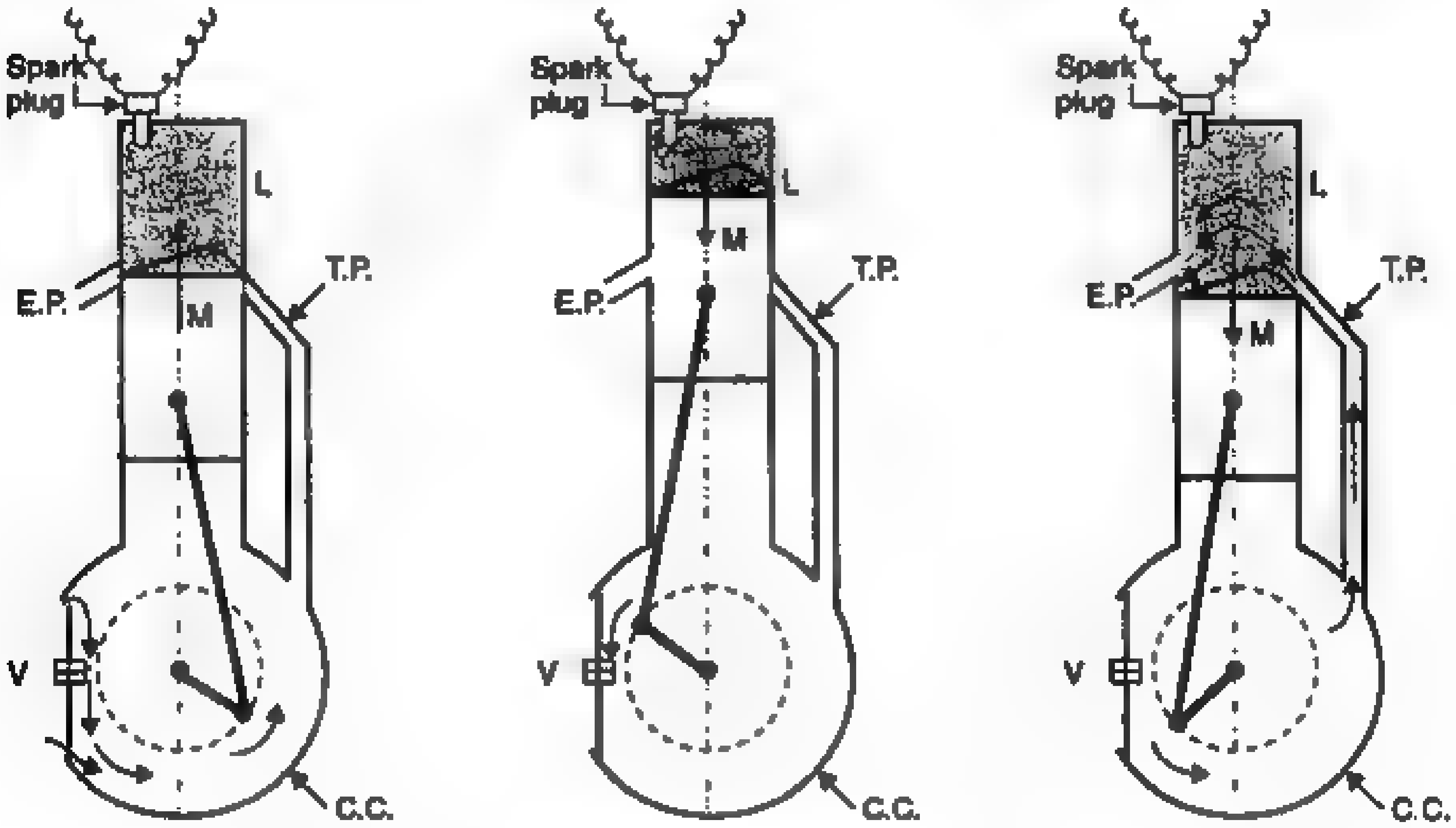


Fig. 2.43. Two stroke cycle engine.

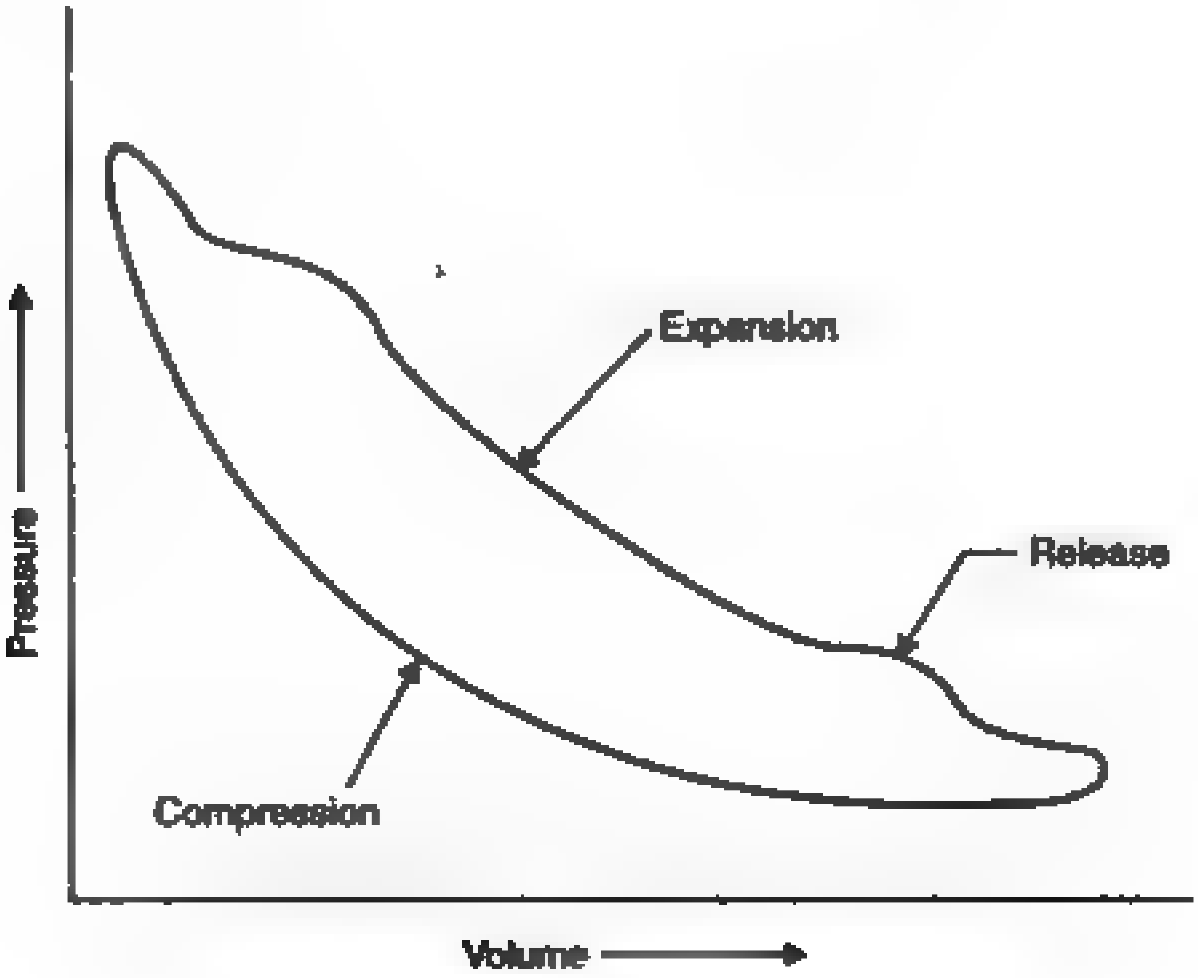


Fig. 2.44. p-V diagram for a two stroke cycle engine.

Fig. 2.44 shows the p-V diagram for a two stroke cycle engine. It is only for the main cylinder or the top side of the piston. Fig. 2.45 shows self-explanatory port timing diagram for a two stroke cycle engine.

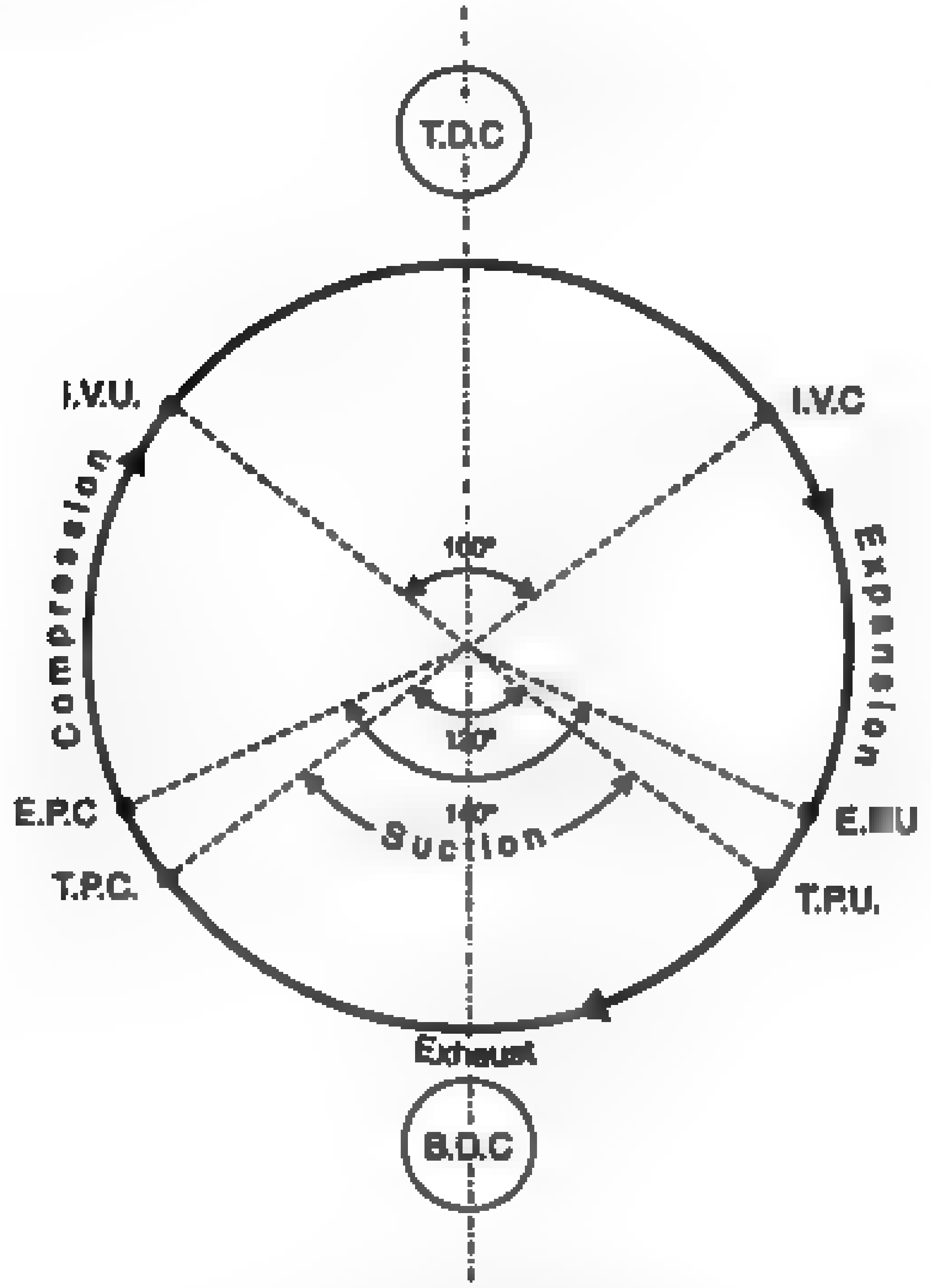


Fig. 2.45. Port timing diagram.

In a two stroke Diesel cycle engine all the operations are the same as in the spark ignition (Otto cycle) engine with the differences ; firstly in this case, only air is admitted into cylinder instead of air fuel mixture and secondly fuel injector is fitted to supply the fuel instead of a sparking plug.

2.13. INTAKE FOR COMPRESSION IGNITION ENGINES

- ***The compression ignition (C.I.) engines are operated unthrottled with engine speed and power controlled by the amount of fuel injected during each cycle.*** **This allows for high volumetric efficiency at all speeds,** ***with the intake system designed for very little flow restriction of the incoming air.*** **Further raising the volumetric efficiency is the fact that no fuel is added until late in compression stroke, after air intake is fully completed. In addition many C.I. engines are** ***turbocharged, which enhances air intake even more.***

- The addition of fuel is made late in the compression stroke, starting somewhere around 20° before T.D.C. Injectors mounted in the cylinder head inject directly into the combustion chamber, where self ignition occurs due to the high temperature of the air caused by compression heating.
- It is important that fuel with the correct cetane number be used in an engine so that self-ignition initiates the start of combustion at the proper cycle position.
- For C.I. engines, the injection pressure must be much higher than that required for S.I. engines. The cylinder pressure into which the fuel is first injected is very high near the end of the compression stroke, due to high compression ratio of C.I. engines. By the time the final fuel is injected, peak pressure during combustion is being experienced. *Pressure must be high enough so that fuel spray will penetrate across the entire combustion chamber.* Injection pressures of 200 bar to 2000 bar are common *with average fuel droplet size generally decreasing with increasing pressure.* Orifice hole size of injectors is typically in the range of 0.2 to 1.0 mm diameter.
- The mass flow rate of fuel ($\dot{m}_f$) through an injector, during injection, is given by the relation :

$$\dot{m}_f = C_D A_n \sqrt{2\rho_f \Delta p} \quad ...(2.1)$$

The total mass of fuel (m_f) injected into one cylinder during one cycle is given as :

$$m_f = C_D A_n \sqrt{2\rho_f \Delta p} \; (\Delta\theta/360\ N) \quad ...(2.2)$$

where, C_D = Discharge co-efficient of injector,
A_n = Flow area of nozzle orifice(s),
ρ_f = Density of fuel,
Δp = Pressure differential across injector,
$\Delta\theta$ = Crank angle through which injection takes place (in degrees), and
N = Engine speed.

Again, $$p_{inj.} \propto \Delta p \quad ...(2.3)$$
and, $$p_{inj.} \propto N^2 \quad ...(2.4)$$

(To ensure that the crank angle of rotation through which injection takes place is almost constant for all speeds)

— *Large engines must have very high injection pressure and high spray velocity.*

— *For optimum fuel viscosity and spary penetration, it is important to have fuel at the correct temperature.*

Often engines are equipped with temperature sensors and means of heating or cooling the incoming fuel. *Many large truck engines are equipped with heated fuel filters. This allows the use of cheaper fuel that has less viscosity control.*

— *In small engines more costly, lower viscosity fuel is required.*

2.14. COMPARISON OF FOUR STROKE AND TWO STROKE CYCLE ENGINES

S.No.	*Aspects*	*Four Stroke Cycle Engines*	*Two Stroke Cycle Engines*
1.	*Completion of cycle*	The cycle is completed in *four strokes of the piston* or in *two revolutions of the crankshaft*. Thus one power stroke is obtained in every two revolutions of the crankshaft.	The cycle is completed in *two strokes of the piston* or in *one revolution of the crankshaft*. Thus one power stroke is obtained in each revolution of the crankshaft.

S.No.	Aspects	Four Stroke Cycle Engines	Two Stroke Cycle Engines
2.	*Flywheel required -heavier or lighter*	Because of the above turning-movement is not so uniform and hence *heavier* flywheel is needed.	More uniform turning movement and hence *lighter* flywheel is needed.
3.	*Power produced for same size of engine*	Again because of one power stroke for two revolutions, power produced for same size of engine is *small* or for the same power the engine is heavy and bulky.	Because of one power stroke for one revolution, power produced for same size of engine is *more* (theoretically twice, actually about 1.3 times) or for the same power the engine is light and compact.
4.	*Cooling and lubrication requirements*	Because of one power stroke in two revolutions *lesser* cooling and lubrication requirements. Lesser rate of wear and tear.	Because of one power stroke in one revolution *greater* cooling and lubrication requirement. Great rate of wear and tear.
5.	*Valve and valve mechanism*	The four stroke engine *contains* valve [redacted]	Two stroke engines have *no* valves but only ports (some two stroke engines are fitted with conventional exhaust valves).
6.	*Initial cost*	Because of the heavy weight and complication of valve mechanism, *higher* is the initial cost.	Because of light weight and simplicity due to absence of valve mechanism, *cheaper* in initial cost.
7.	*Volumetric efficiency*	Volumetric efficiency *more* due to more time of induction.	Volumetric efficiency *less* due to lesser time for induction.
8.	*Thermal and part-load efficiencies*	Thermal efficiency higher, part load effitiency better than two stroke cycle engine.	Thermal efficiency lower, part load efficiency lesser than four stroke cycle engine.
9.	*Applications*	Used where efficiency is important ; in *cars, buses, trucks, tractors, industrial engines, aeroplane, power generators etc.*	In two stroke petrol engine some fuel is exhausted during scavenging. Used where (*a*) *low cost*, and (*b*) *compactness and light weight important.* Two stroke (air cooled) petrol engines used in very small sizes only, *lawn movers, scooters motor cycles* (lubricating oil mixed with petrol). Two stroke diesel engines used in *very large sizes* more than 60 cm bore, for *ship propulsion* because of low weight and compactness.

2.15. COMPARISON OF SPARK IGNITION (S.I.) AND COMPRESSION IGNITION (C.I.) ENGINES

S.No.	Aspects	S.I. engines	C.I. engines
1.	*Thermodynamic cycle*	Otto cycle	Diesel cycle ...... For slow speed engines Dual cycle ...... For high speed engines
2.	*Fuel used*	Petrol	Diesel.

S.No.	*Aspects*	*S.I. engines*	*C.I. engines*
3.	*Air-fuel ratio*	10 : 1 to 20 : 1	18 : 1 to 100 : 1.
4.	*Compression ratio*	upto 11 ; Average value 7 to 9 ; Upper limit of compression ratio fixed by *anti-knock quality of fuel.*	12 to 24 ; Average value 15 to 18 ; Upper limit of compression ratio is limited by *thermal and mechanical stresses.*
5.	*Combustion*	Spark ignition	Compression ignition.
6.	*Fuel supply*	By carburettor ...... cheap method	By injection ...... expensive method.
7.	*Operating pressure* (*i*) Compression pressure (*ii*) Maximum pressure	7 bar to 15 bar 45 bar to 60 bar	30 bar to 50 bar 60 bar to 120 bar.
8.	*Operating speed*	High speed : 2000 to 6000 r.p.m.	Low speed . 400 r p.m. Medium speed : 400 to 1200 r.p.m. High speed : 1200 to 3500 r p.m.
9.	*Control of power*	Quantity governing ...... by throttle	Quality governing ..... by rack.
10.	*Calorific value*	44 MJ/kg	42 MJ/kg.
11.	*Cost of running*	high	low.
12.	*Maintenance cost*	Minor maintenance required	Major overall required but less frequently.
13.	*Supercharging*	Limited by *detonation.* Used only in *aircraft engines.*	Limited by *blower power and mechanical and thermal stresses. Widely used.*
14.	*Two stroke operation*	*Less suitable,* fuel loss in scavenging. But small two stroke engines are used in mopeds, scooters and motorcycles due to their *simplicity and low cost.*	No fuel loss in scavenging. *More suitable.*
15.	*High powers*	No	Yes.
16.	*Distribution of fuel*	A/F ratio is not optimum in multi-cylinder engines.	Excellent distribution of fuel in multi-cylinder engines.
17.	*Starting*	Easy, low cranking effort.	Difficult, high cranking effort.
18.	*Exhaust gas temperature*	High, due to low thermal efficiency.	Low, due to high thermal efficiency.
19.	*Weight per unit power*	Low (0.5 to 4.5 kg/kW).	High (3.3 to 13.5 kg/kW).
20.	*Initial capital cost*	Low	High due to heavy weight and study construction, costly construction, 1.25-1.5 times.
21.	*Noise and vibration*	Less	More idle noise problem.
22.	*Uses*	Mopeds, scooters, motorcycles, simple engine passenger cars, aircrafts etc.	Buses, trucks locomotives, tractors, earth moving machinery and stationary generating plants.

2.16. COMPARISON BETWEEN A PETROL ENGINE AND A DIESEL ENGINE

S.No.	*Petrol engine*	*Diesel engine*
1.	Air petrol mixture is sucked in the engine cylinder during suction stroke.	Only air is sucked during suction stroke.
2.	Spark plug is used.	Employs an injector.
3.	Power is produced by spark ignition.	Power is produced by compression ignition.
4.	Thermal efficiency up to 25%.	Thermal efficiency up to 40%.
5.	Occupies less space.	Occupies more space.
6.	More running cost.	Less running cost.
7.	Light in weight.	Heavy in weight.
8.	Fuel (Petrol) costlier.	Fuel (Diesel) cheaper.
9.	Petrol being volatile is dangerous.	Diesel is non-dangerous as it is non-volatile.
10.	Pre-ignition possible.	Pre-ignition not possible.
11.	Works on Otto cycle.	Works on Diesel cycle.
12.	Less dependable.	More dependable.
13.	Used in *cars* and *motor cycles*.	Used in heavy duty vehicles like *trucks, buses* and *heavy machinery*.

2.17. HOW TO TELL A TWO STROKE CYCLE ENGINE FROM A FOUR STROKE CYCLE ENGINE ?

S.No.	*Distinguishing features*	*Four stroke cycle engine*	*Two stroke cycle engine*
1.	*Oil sump and oil-filter plug*	It has an oil sump and oil-filter plug.	It does not have oil sump and oil-filter plug.
2.	*Oil drains etc.*	It requires oil drains and refills periodically, just an automobile do.	In this type of engine, the oil is added to the gasoline so that a mixture of gasoline and oil passes through the carburettor and enters the crankcase with the air
3.	*Location of muffler (exhaust silencer)*	It is installed at the head end of the cylinder at the exhaust valve location.	It is installed towards the middle of the cylinder, at the exhaust port location.
4.	Name plate	If the name plate mentions the type of oil and the crankcase capacity, or similar data, it is a four stroke cycle engine.	If the name plate tells to mix oil with the gasoline, it is a two stroke cycle engine.

HIGHLIGHTS

1. Any type of engine or machine which derives heat energy from the combustion of fuel or any other source and converts this energy into mechanical work is termed as a **heat engine**.
2. The function of a carburettor is to atomise and meter the liquid fuel and mix it with air as it enters the injection system of the engine maintaining under all conditions of operation fuel air proportion approximate to those conditions.
3. The two basic ignition systems in current use are :
   (*i*) Battery or coil ignition system
   (*ii*) Magneto ignition system.
4. Following are the methods of governing I.C. engines :
   (*i*) Hit and miss method (*ii*) Quality governing
   (*iii*) Quantity governing.
5. Pre-ignition is the premature combustion which starts before the application of spark. Overheated spark plugs and exhaust valves which are the main causes of pre-ignition should be carefully avoided in engines.
6. A very sudden rise to pressure during combustion accompanied by metallic hammer like sound is called **detonation**. The region in which detonation occurs is farthest removed from the sparking plug, and is named the 'detonation zone' and even with severe detonation this zone is rarely more than that one quarter the clearance volume.
7. The **octane number** is the percentage of octane in the mixture [of iso-octane (high rating) and normal heptane (low rating), by volume] which knocks under the same conditions as the fuel.
8. **Delay period or ignition lag** is the time immediately following injection of fuel during which the ignition process is being initiated and the pressure does not rise beyond the value it would have due to compression of air.
9. Higher the cetane rating of the fuel lesser is the propensity for diesel knock. In general a high octane value implies a low cetane value.
10. The purpose of **supercharging** is to raise the volumetric efficiency above that value that which can be obtained by normal aspiration. Supercharging of petrol engines, because of its poor fuel economy, is not very popular and is used only when a large amount of power is needed or when more power is needed to compensate altitude loss.
11. **Dissociation** refers to disintegration of burnt gases at high temperatures. It is a reversible process and increases with temperature. Dissociation, in general, causes a loss of power and efficiency.
12. **Performance of I.C. engines.** Some important relations :

(*i*) Indicated power (I.P.) $= \dfrac{np_m LANk \times 10}{6}$ kW

(*ii*) Brake Power (B.P.) $= \dfrac{(W - S)\,\pi(D_b + d)N}{60 \times 1000}$ kW or $\left(= \dfrac{2\pi NT}{60 \times 1000}\,\text{kW}\right)$

(*iii*) Mechanical efficiency, $\eta_{mech.} = \dfrac{\text{B.P.}}{\text{I.P.}}$

(*iv*) Thermal efficiency (indicated), $\eta_{th.(i)} = \dfrac{\text{I.P.}}{\dot{m}_f \times C}$

and thermal efficiency (brake), $\eta_{th.(b)} = \dfrac{\text{B.P.}}{\dot{m}_f \times C}$

where $\dot{m}_f$ = mass of fuel used in kg/sec.

(*v*) $\eta_{relative} = \dfrac{\eta_{thermal}}{\eta_{air\text{-}standard}}$

(*vi*) Measurement of air consumption by **air box method** :

Volume of air passing through the orifice, $V_a = 840 AC_d \sqrt{\frac{h_w}{\rho_a}}$

and mass of air passing through the orifice,

$$m_a = 0.066\, C_d \times d^2 \sqrt{h_w \rho_a} \text{ kg/min}$$

where, A = Area of orifice, m^2
d = Diameter of orifice, cm
h_w = Head of water in 'cm' causing the flow
ρ_a = Density of air in kg/m^3 under atmospheric conditions.

OBJECTIVE TYPE QUESTIONS

Choose the correct answer :

1. In a four stroke cycle engine, the four operations namely suction, compression, expansion and exhaust are completed in the number of revolutions of crank shaft equal to
(*a*) four (*b*) three
(*c*) two (*d*) one.
2. In a two stroke cycle engine, the operations namely suction, compression, expansion and exhaust are completed in the number of revolutions of crank shaft equal to
(*a*) four (*b*) three
(*c*) two (*d*) one.
3. In a four stroke cycle S.I. engine the cam shaft runs
(*a*) at the same speed as crank shaft (*b*) at half the speed of crank shaft
(*c*) at twice the speed of crank shaft (*d*) at any speed irrespective of crank shaft speed.
4. The following is an S.I. engine
(*a*) Diesel engine (*b*) Petrol engine
(*c*) Gas engine (*d*) none of the above.
5. The following is C.I. engine
(*a*) Diesel engine (*b*) Petrol engine
(*c*) Gas engine (*d*) none of the above.
6. In a four stroke cycle petrol engine, during suction stroke
(*a*) only air is sucked in (*b*) only petrol is sucked in
(*c*) mixture of petrol and air is sucked in (*d*) none of the above.
7. In a four stroke cycle diesel engine, during suction stroke
(*a*) only air is sucked in (*b*) only fuel is sucked in
(*c*) mixture of fuel and air is sucked in (*d*) none on the above.
8. The two stroke cycle engine has
(*a*) one suction valve and one exhaust valve operated by one cam
(*b*) one suction valve and one exhaust valve operated by two cams
(*c*) only ports covered and uncovered by piston to effect charging and exhausting
(*d*) none of the above.
9. For same output, same speed and same compression ratio the thermal efficiency of a two stroke cycle petrol engine as compared to that for four stroke cycle petrol engine is
(*a*) more (*b*) less
(*c*) same as long as compression ratio is same (*d*) same as long as output is same.
10. The ratio of brake power to indicated power of an I.C. engine is called
(*a*) mechanical efficiency (*b*) thermal efficiency
(*c*) volumetric efficiency (*d*) relative efficiency.

ANSWERS

1. (*c*) 2. (*d*) 3. (*b*) 4. (*b*) 5. (*a*) 6. (*c*) 7. (*a*)
8. (*c*) 9. (*b*) 10. (*a*).

THEORETICAL QUESTIONS

1. Name the two general classes of combustion engines and state how do they basically differ in principle ?
2. Discuss the relative advantages and disadvantages of internal combustion and external combustion engines.
3. What are the two basic types of internal combustion engines ? What are the fundamental differences between the two ?
4. What is the function of a governor ? Enumerate the types of governors and discuss with a neat sketch the Porter governor.
5. Differentiate between a flywheel and a governor.
6. (*a*) State the function of a carburettor in a petrol engine.
   (*b*) Describe a simple carburettor with a neat sketch and also state its limitations.
7. Explain with neat sketches the construction and working of the following :
   (*i*) Fuel pump (*ii*) Injector.
8. Explain the following terms as applied to I.C. engines :
   Bore, stroke, T.D.C., B.D.C., clearance volume, swept volume, compression ratio and piston speed.
9. Explain with suitable sketches the working of a four stroke otto engine.
10. Discuss the difference between ideal and actual valve timing diagrams of a petrol engine.
11. In what respects four stroke diesel cycle (compression ignition) engine differs from four stroke cycle spark ignition engine ?
12. Discuss the difference between theoretical and actual valve timing diagrams of a diesel engine ?
13. What promotes the development of two stroke engines ? What are the two main types of two stoke engines ?
14. Describe with a suitable sketch the two stroke cycle spark ignition (SI) engine. How its indicator diagram differs from that of four stroke cycle engine ?
15. Compare the relative advantages and disadvantages of four stroke and two stroke cycle engines.

3

Air Standard Cycles

3.1. Definition of a cycle. 3.2. Air standard efficiency. 3.3. The Carnot cycle. 3.4. Constant volume or Otto cycle. 3.5. Constant pressure or Diesel cycle. 3.6. Dual combustion cycle. 3.7. Comparison of Otto, Diesel and Dual combustion cycles—Efficiency versus compression ratio—For the same compression ratio and the same heat input—For constant maximum pressure and heat supplied. 3.8. Atkinson cycle. 3.9. Ericsson cycle. 3.10. Brayton cycle. 3.11. Stirling cycle. 3.12. Miller cycle. 3.13. Lenoir cycle—Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

3.1. DEFINITION OF A CYCLE

A **cycle** is defined as a *repeated series of operations occuring in a certain order*. It may be repeated by repeating the processes in the same order. The cycle may be of imaginary perfect engine or actual engine. The former is called **ideal cycle** and the latter **actual cycle**. In ideal cycle all accidental heat losses are prevented and the working substance is assumed to behave like a perfect working substance.

3.2. AIR STANDARD EFFICIENCY

To compare the effects of different cycles, it is of paramount importance that the effect of the calorific value of the fuel is altogether eliminated and this can be achieved by considering air (which is assumed to behave as a perfect gas) as the working substance in the engine cylinder. *The efficiency of engine using air as the working medium is known as an* **"Air standard efficiency"**. This efficiency is oftenly called **ideal efficiency.**

The actual efficiency of a cycle is always *less* than the air-standard efficiency of that cycle under ideal conditions. This is taken into account by introducing a new term **"Relative efficiency"** which is defined as :

$$\eta_{relative} = \frac{\text{Actual thermal efficiency}}{\text{Air standard efficiency}} \quad ...(3.1)$$

The analysis of all air standard cycles is based upon the following *assumptions* :

Assumptions :

1. The gas in the engine cylinder is a perfect gas *i.e.*, it obeys the gas laws and has constant specific heats.
2. The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures *i.e.*, the molecular weight of cylinder gas is 29.

$$c_p = 1.005 \text{ kJ/kg-K}, \; c_v = 0.718 \text{ kJ/kg-K}.$$

3. The compression and expansion processes are adiabatic and they take place without internal friction, *i.e.*, these processes are isentropic.
4. No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in contact with cylinder at appropriate points during the process.

5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

3.3. THE CARNOT CYCLE

This cycle has the *highest possible efficiency* and consists of four simple operations namely,

(*a*) Isothermal expansion

(*b*) Adiabatic expansion

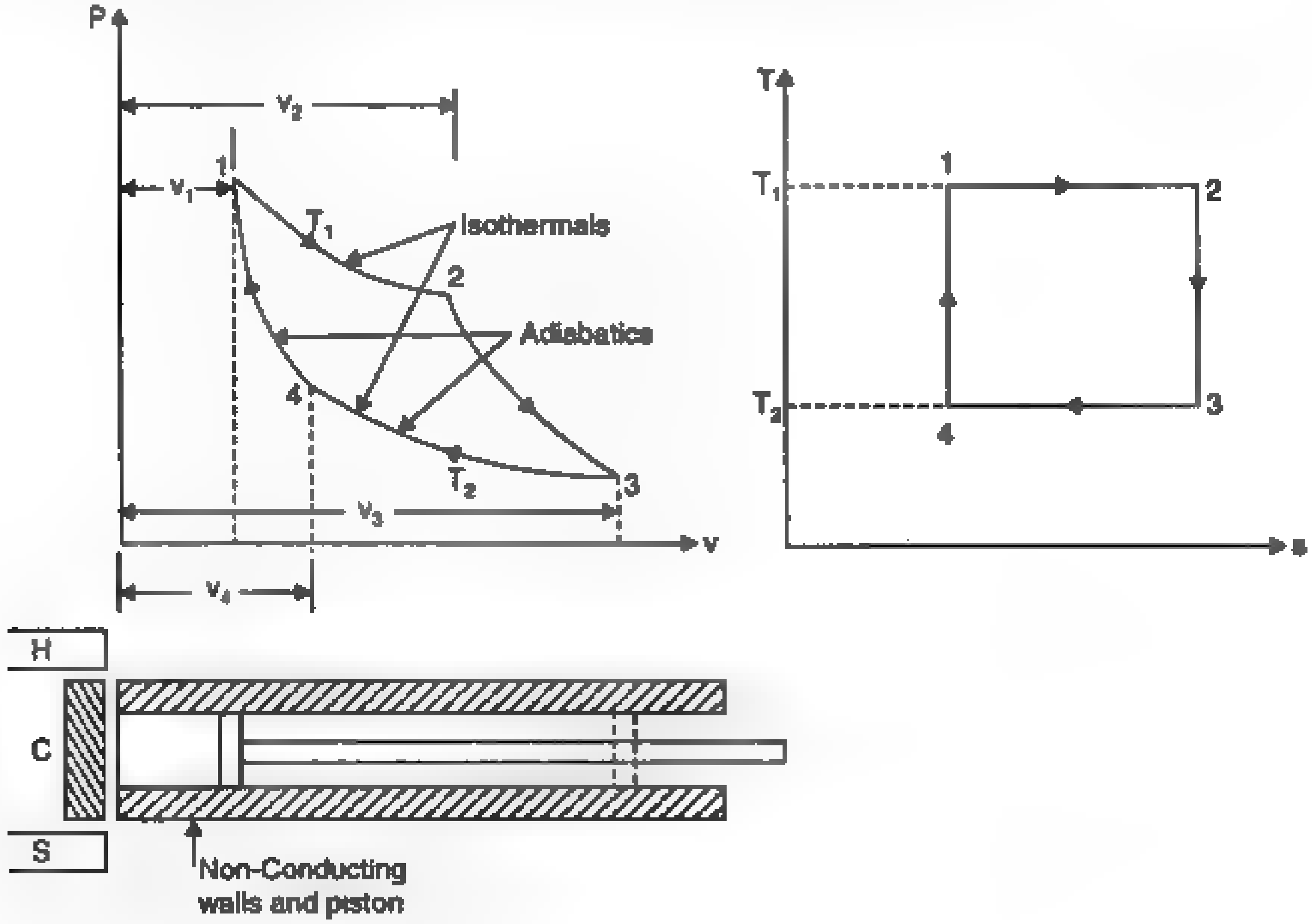
(*c*) Isothermal compression

(*d*) Adiabatic compression.

The condition of the Carnot cycle may be imagined to occur in the following way :

One kg of a air is enclosed in the cylinder which (except at the end) is made of perfect non-conducting material. A source of heat '*H*' is supposed to provide unlimited quantity of heat, non-conducting cover 'C' and a sump '*S*' which is of infinite capacity so that its temperature remains unchanged irrespective of the fact how much heat is supplied to it. The temperature of source *H* is T_1 and the same is of the working substance. The working substance while rejecting heat to sump '*S*' has the temperature T_2 *i.e.*, the same as that of sump *S*.

Following are the *four stages* of the Carnot cycle. Refer Fig. 3.1 (*a*).



(*a*) Four stages of carnot cycle (*b*) T-s diagram

Fig. 3.1

Stage (1). Line 1-2 [Fig. 3.1 (*a*)] represents the isothermal expansion which takes place at temperature T_1 when source of heat H is applied to the end of cylinder. Heat supplied in this case is given by $RT_1 \log_e r$ and where r is the ratio of expansion.

Stage (2). Line 2-3 represents the application of non-conducting cover to the end of the cylinder. This is followed by the adiabatic expansion and the temperature falls from T_1 to T_2.

Stage (3). Line 3-4 represents the isothermal compression which takes place when sump '*S*' is applied to the end of cylinder. Heat is rejected during this operation whose value is given by $RT_2 \log_e r$ where r is the ratio of compression.

Stage (4). Line 4-1 represents repeated application of non-conducting cover and adiabatic compression due to which temperature increases from T_2 to T_1.

It may be noted that ratio of expansion during isothermal 1-2 and ratio of compression during isothermal 3-4 must be equal to get a closed cycle.

Fig. 3.1 (*b*) represents the Carnot cycle on *T-s* coordinates.

Now according to law of conservation of energy,

Heat supplied = Work done + Heat rejected

Work done = Heat supplied – Heat rejected

$$= RT_1 \,.\, \log_e r - RT_2 \log_e r$$

$$\text{Efficiency of cycle} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{R \log_e r\,(T_1 - T_2)}{RT_1 \,.\, \log_e r}$$

$$= \frac{T_1 - T_2}{T_1} \qquad \text{...(3.2)}$$

From this equation, it is quite obvious that if temperature T_2 decreases, efficiency increases and it becomes 100% if T_2 becomes absolute zero which, of course is impossible to attain. Further more *it is not possible to produce an engine that should work on Carnot's cycle as it would necessitate the piston to travel very slowly during first portion of the forward stroke (isothermal expansion) and to travel more quickly during the remainder of the stroke (adiabatic expansion) which however is not practicable.*

Example 3.1. *A Carnot engine working between 400°C and 40°C produces 130 kJ of work. Determine :*

(*i*) *The engine thermal efficiency.*

(*ii*) *The heat added.*

(*iii*) *The entropy changes during heat rejection process.*

Solution. Temperature, $T_1 = T_2 = 400 + 273 = 673$ K

Temperature, $T_3 = T_4 = 40 + 273 = 313$ K

Work produced, $W = 130$ kJ.

(*i*) **Engine thermal efficiency, $\eta_{th.}$:**

$$\eta_{th.} = \frac{673 - 313}{673} = \mathbf{0.535 \text{ or } 53.5\%.} \quad \textbf{(Ans.)}$$

(*ii*) **Heat added :**

$$\eta_{th.} = \frac{\text{Work done}}{\text{Heat added}}$$

i.e.,

$$0.535 = \frac{130}{\text{Heat added}}$$

$$\therefore \quad \text{Heat added} = \frac{130}{0.535} = \mathbf{243 \text{ kJ.}} \quad \textbf{(Ans.)}$$

(*iii*) **Entropy change during the heat rejection process, $(S_3 - S_4)$:**

Heat rejected = Heat added – Work done

= 243 – 130 = 113 kJ

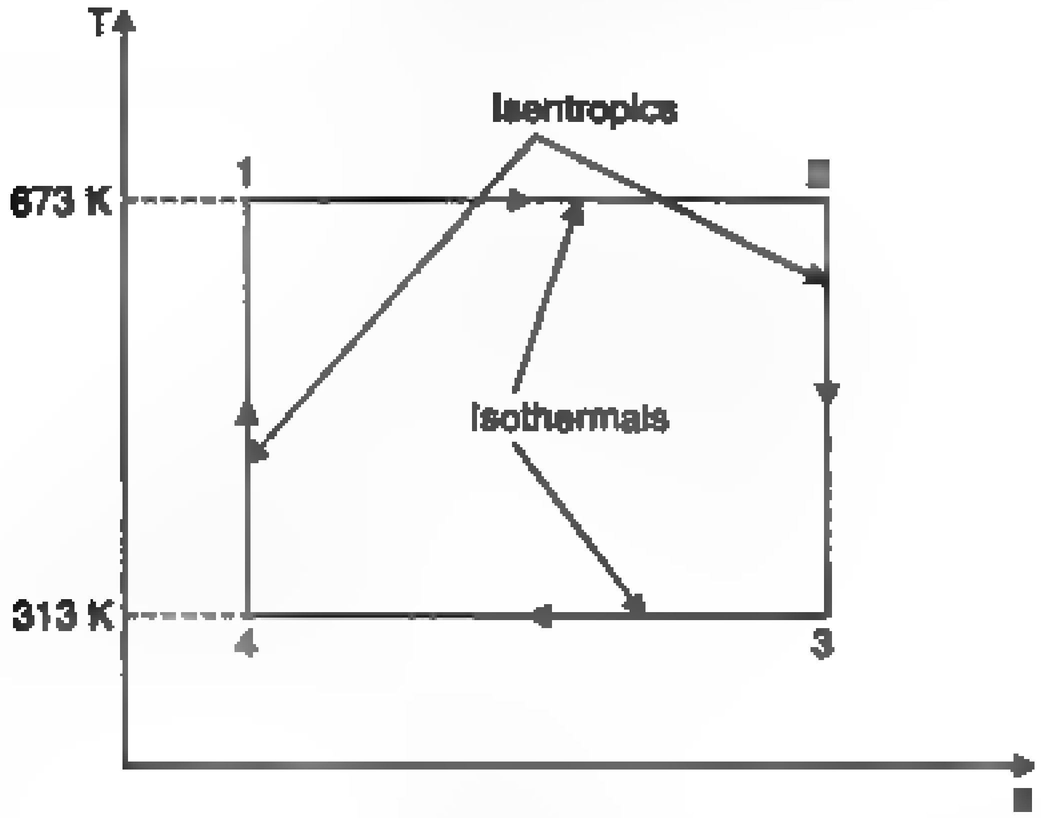


Fig. 3.2

Heat rejected $= T_3 (S_3 - S_4) = 113$

$\therefore \quad (S_3 - S_4) = \dfrac{113}{T_3} = \dfrac{113}{313} =$ **0.361 kJ/K. (Ans.)**

Example 3.2. *0.5 kg of air (ideal gas) executes a Carnot power cycle having a thermal efficiency of 50 per cent. The heat transfer to the air during the isothermal expansion is 40 kJ. At the beginning of the isothermal expansion the pressure is 7 bar and the volume is 0.12 m³. Determine :*

(i) The maximum and minimum temperatures for the cycle in K ;

(ii) The volume at the end of isothermal expansion in m³ ;

(iii) The heat transfer for each of the four processes in kJ.

For air c_v = 0.721 kJ/kg K, and c_p = 1.008 kJ/kg K. **(U.P.S.C. 1993)**

Solution. Refer Fig. 3.3. *Given* : m = 0.5 kg ; η_{th} = 50% ; Heat transfered during isothermal expansion = 40 kJ ; p_1 = 7 bar, V_1 = 0.12 m³ ; c_v = 0.721 kJ/kg K ; c_p = 1.008 kJ/kg K.

(*i*) **The maximum and minimum temperatures, T_1, T_2 :**

$$p_1 V_1 = mRT_1$$

$$7 \times 10^5 \times 0.12 = 0.5 \times 287 \times T_1$$

$\therefore$ Maximum temperature, $T_1 = \dfrac{7 \times 10^5 \times 0.12}{0.5 \times 287} =$ **585.4 K. Ans.**

$$\eta_{cycle} = \frac{T_1 - T_2}{T_1} \Rightarrow 0.5 = \frac{585.4 - T_2}{585.4}$$

$\therefore$ Minimum temperature, $T_2 = 585.4 - 0.5 \times 585.4 =$ **292.7 K. (Ans.)**

(*ii*) **The volume at the end of isothermal expansion V_2 :**

Heat transferred during isothermal expansion

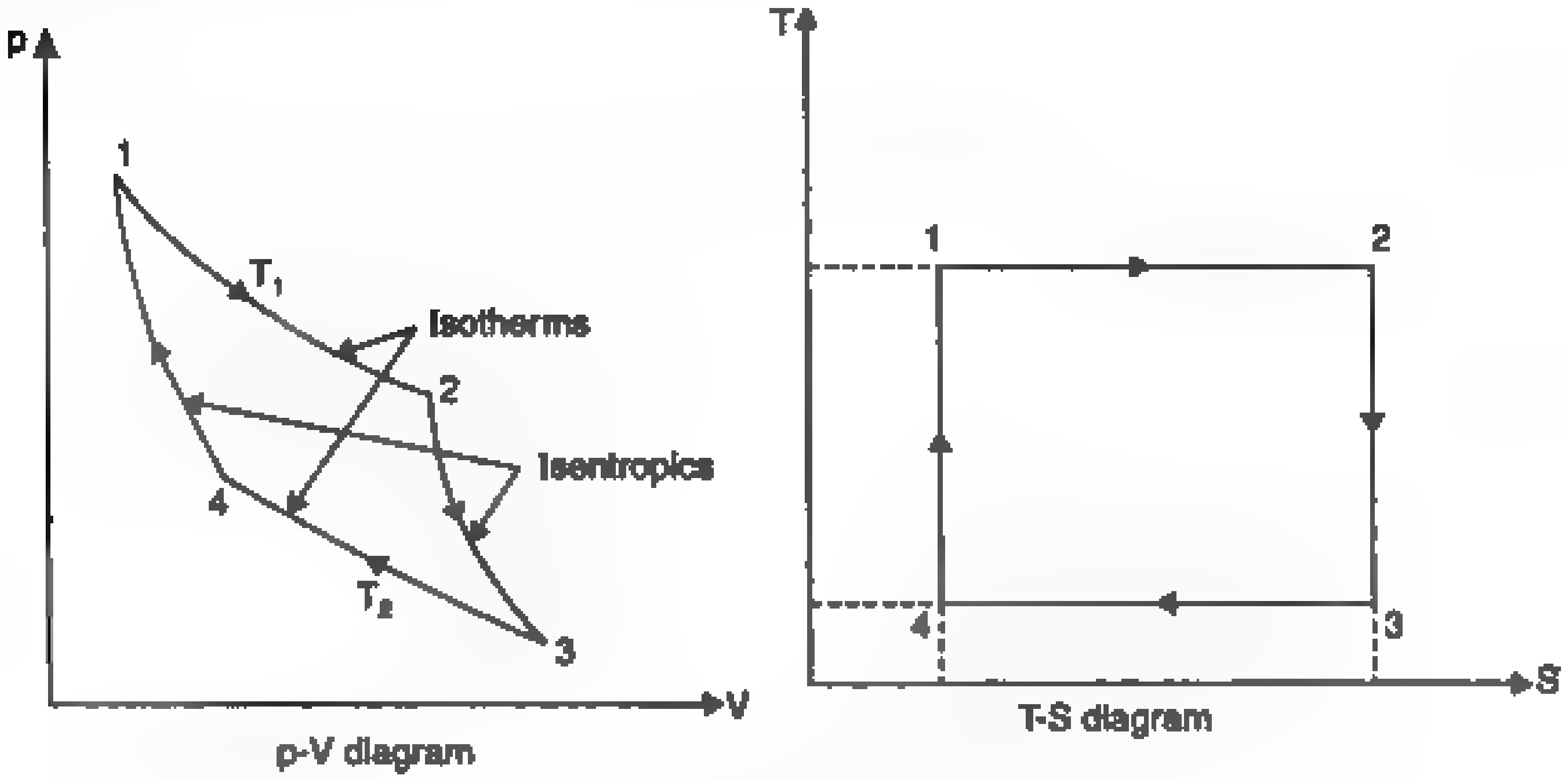


Fig. 3.3. Carnot cycle.

$$= p_1V_1 \ln(r) = mRT_1 \ln\left(\frac{V_2}{V_1}\right) = 40 \times 10^3 \quad \text{......(Given)}$$

or $$0.5 \times 287 \times 585.4 \ln\left(\frac{V_2}{0.12}\right) = 40 \times 10^3$$

or $$\ln\left(\frac{V_2}{0.12}\right) = \frac{40 \times 10^3}{0.5 \times 287 \times 585.4} = 0.476$$

or $$V_2 = 0.12 \times (e)^{0.476} = \mathbf{0.193\ m^3}. \quad \textbf{(Ans.)}$$

(*iii*) **The heat transfer for each of the four processes :**

Process	*Classification*	*Heat transfer*
1—2	Isothermal expansion	40 kJ
2—3	Adiabatic reversible expansion	zero
3—4	Isothermal compression	– 40 kJ
4—1	Adiabatic reversible compression	zero. **(Ans.)**

☞**Example 3.3.** *In a Carnot cycle, the maximum pressure and temperature are limited to 18 bar and 410°C. The ratio of isentropic compression is 6 and isothermal expansion is 1.5. Assuming the volume of the air at the beginning of isothermal expansion as 0.18 m³, determine :*

(*i*) *The temperature and pressures at main points in the cycle.*

(*ii*) *Change in entropy during isothermal expansion.*

(*iii*) *Mean thermal efficiency of the cycle.*

(*iv*) *Mean effective pressure of the cycle*

(*v*) *The theoretical power if there are 210 working cycles per minute.*

Solution. Refer Fig. 3.4.

Maximum pressure, $p_1 = 18$ bar

Maximum temperature, $T_1 = (T_2) = 410 + 273 = 683$ K

Ratio of isentropic (or adiabatic) compression, $\dfrac{V_4}{V_1} = 6$

Ratio of isothermal expansion, $\dfrac{V_2}{V_1} = 1.5.$

Volume of the air at the beginning of isothermal expansion, $V_1 = 0.18$ m^3.

(*i*) **Temperatures and pressures at the main points in the cycle :**

For the *isentropic process* 4-1

$$\frac{T_1}{T_4} = \left(\frac{V_4}{V_1}\right)^{\gamma - 1} = (6)^{1.4 - 1} = (6)^{0.4} = 2.05$$

$$\therefore \qquad T_4 = \frac{T_1}{2.05} = \frac{683}{2.05} = 333.2 \text{ K} = T_3$$

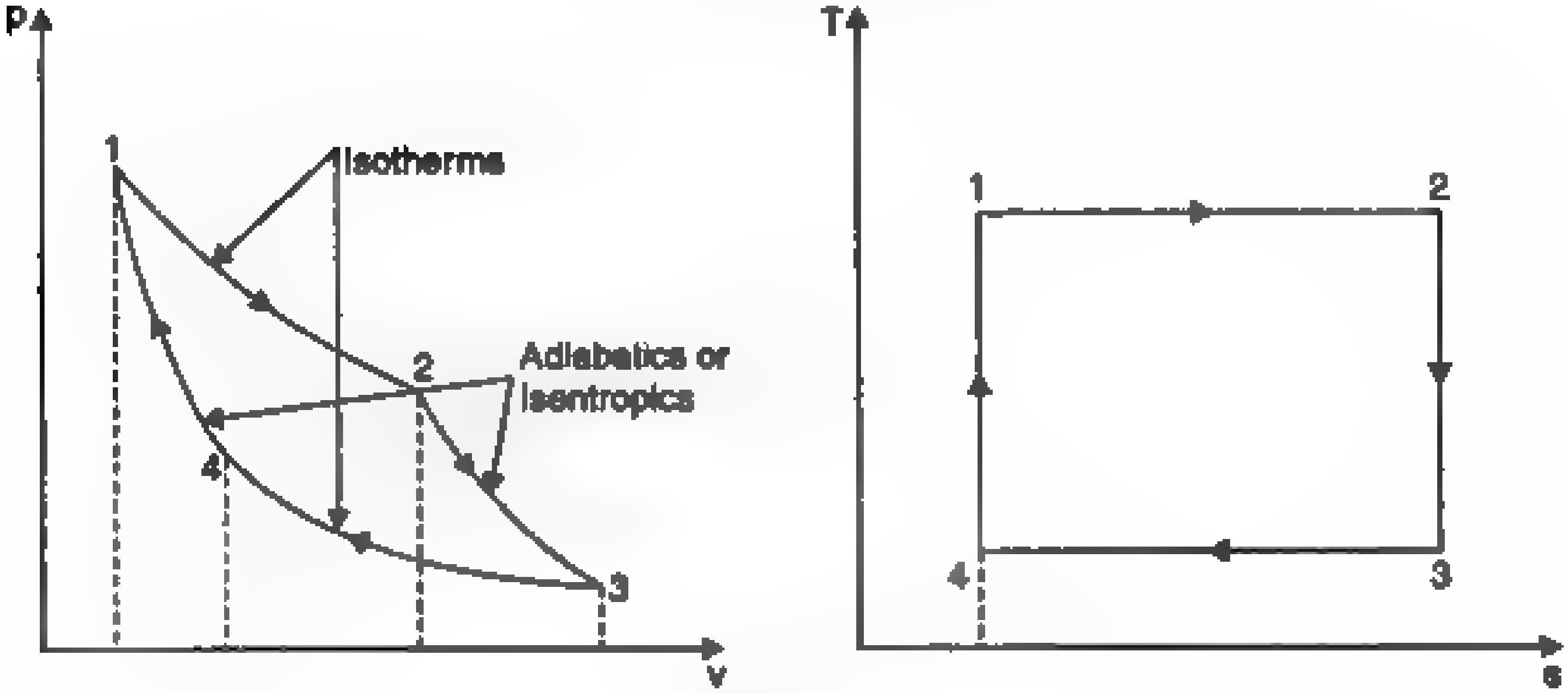


Fig. 3.4

Also,
$$\frac{p_1}{p_4} = \left(\frac{V_4}{V_1}\right)^{\gamma} = (6)^{1.4} = 12.29$$

$$\therefore \qquad p_4 = \frac{p_1}{12.29} = \frac{18}{12.29} = 1.46 \text{ bar}$$

For the *isothermal process* 1-2

$$p_1V_1 = p_2V_2$$

$$p_2 = \frac{p_1V_1}{V_2} = \frac{18}{1.5} = 12 \text{ bar}$$

For *isentropic process* 2-3, we have

$$p_2V_2^{\gamma} = p_3V_3^{\gamma}$$

$$p_3 = p_2 \times \left(\frac{V_2}{V_3}\right)^{\gamma} = 12 \times \left(\frac{V_1}{V_4}\right)^{\gamma} \qquad \left[\because \frac{V_4}{V_1} = \frac{V_3}{V_2}\right]$$

$$= 12 \times \left(\frac{1}{6}\right)^{1.4} = \mathbf{0.97\ bar.}\quad \textbf{(Ans.)}$$

Hence

$$\left.\begin{array}{ll} p_1 = \mathbf{18\ bar} & T_1 = T_2 = \mathbf{683\ K} \\ p_2 = \mathbf{12\ bar} & \\ p_3 = \mathbf{0.97\ bar} & T_3 = T_4 = \mathbf{333.2\ K} \\ p_4 = \mathbf{1.46\ bar} & \end{array}\right\} \textbf{(Ans.)}$$

(*ii*) **Change in entropy :**

Change in entropy during isothermal expansion,

$$S_2 - S_1 = mR \log_e \left(\frac{V_2}{V_1}\right) = \frac{p_1V_1}{T_1} \log_e \left(\frac{V_2}{V_1}\right) \qquad \left[\because pV = mRT \text{ or } mR = \frac{pV}{T}\right]$$

$$= \frac{18 \times 10^5 \times 0.18}{10^3 \times 683} \log_e (1.5) = \mathbf{0.192\ kJ/K.}\quad \textbf{(Ans.)}$$

(*iii*) **Mean thermal efficiency of the cycle :**

Heat supplied,

$$Q_s = p_1V_1 \log_e \left(\frac{V_2}{V_1}\right)$$
$$= T_1 (S_2 - S_1)$$
$$= 683 \times 0.192 = 131.1 \text{ kJ}$$

Heat rejected,

$$Q_r = p_4V_4 \log_e \left(\frac{V_3}{V_4}\right)$$

$= T_4 (S_3 - S_4)$ because increase in entropy during heat addition is equal to decrease in entropy during heat rejection.

$\therefore$ $Q_r = 333.2 \times 0.192 = 63.97$ kJ

$\therefore$ *Efficiency,*

$$\eta = \frac{Q_s - Q_r}{Q_s} = 1 - \frac{Q_r}{Q_s}$$

$$= 1 - \frac{63.97}{131.1} = \mathbf{0.512 \ \ or \ \ 51.2\%.}\quad \textbf{(Ans.)}$$

(*iv*) **Mean effective pressure of the cycle, p_m :**

The mean effective pressure of the cycle is given by

$$p_m = \frac{\text{Work done per cycle}}{\text{Stroke volume}}$$

$$\frac{V_3}{V_1} = 6 \times 1.5 = 9$$

Stroke volume, $V_s = V_3 - V_1 = 9V_1 - V_1 = 8V_1 = 8 \times 0.18 = 1.44 \text{ m}^3$

$$\therefore \quad p_m = \frac{(Q_s - Q_r) \times J}{V_s} = \frac{(Q_s - Q_r) \times 1}{V_s} \qquad (\because J = 1)$$

$$= \frac{(131.1 - 63.97) \times 10^3}{1.44 \times 10^5} = \mathbf{0.466\ bar.}\quad \textbf{(Ans.)}$$

(*v*) **Power of the engine, P :**

Power of the engine working on this cycle is given by

$$P = (131.1 - 63.97) \times (210/60) = \mathbf{234.9\ kW.} \quad \textbf{(Ans.)}$$

Example 3.4. *A reversible engine converts one-sixth of the heat input into work. When the temperature of the sink is reduced by 70°C, its efficiency is doubled. Find the temperature of the source and the sink.*

Solution. Let T_1 = Temperature of the source (*K*), and

T_2 = Temperature of the sink (*K*).

First case :

$$\frac{T_1 - T_2}{T_1} = \frac{1}{6}$$

i.e., $$6T_1 - 6T_2 = T_1$$

or $$5T_1 = 6T_2 \quad \text{or} \quad T_1 = 1.2T_2 \qquad ...(i)$$

Second case :

$$\frac{T_1 - [T_2 - (70 + 273)]}{T_1} = \frac{1}{3}$$

$$\frac{T_1 - T_2 + 343}{T_1} = \frac{1}{3}$$

$$3T_1 - 3T_2 + 1029 = T_1$$

$$2T_1 = 3T_2 - 1029$$

$$2 \times (1.2T_2) = 3T_2 - 1029 \qquad (\because \ T_1 = 1.2\ T_2)$$

$$2.4T_2 = 3T_2 - 1029$$

or $$0.6T_2 = 1029$$

$\therefore$ $$T_2 = \frac{1029}{0.6} = \mathbf{1715\ K} \quad \text{or} \quad \mathbf{1442°C.} \quad \textbf{(Ans.)}$$

and $$T_1 = 1.2 \times 1715 = \mathbf{2058\ K} \quad \text{or} \quad \mathbf{1785°C.} \quad \textbf{(Ans.)}$$

Example 3.5. *An inventor claims that a new heat cycle will develop 0.4 kW for a heat addition of 32.5 kJ/min. The temperature of heat source is 1990 K and that of sink is 850 K. Is his claim possible ?*

Solution. Temperature of heat source, T_1 = 1990 K

Temperature of sink, T_2 = 850 K

Heat supplied, = 32.5 kJ/min

Power developed by the engine, P = 0.4 kW

The most efficient engine is one that works on Carnot cycle

$\therefore$ $$\eta_{carnot} = \frac{T_1 - T_2}{T_1} = \frac{1990 - 850}{1990} = 0.573 \text{ or } 57.3\%$$

Also, thermal efficiency of the engine,

$$\eta_{th} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{0.4}{(32.5/60)} = \frac{0.4 \times 60}{32.5}$$

$$= 0.738 \text{ or } 73.8\%$$

which is not feasible as no engine can be more efficient than that working on Carnot cycle.

Hence claims of the inventor is **not true. (Ans.)**

Example 3.6. *An ideal engine operates on the Carnot cycle using a perfects gas as the working fluid. The ratio of the greatest to the least volume is fixed and is x : 1, the lower temperature of the cycle is also fixed, but the volume compression ratio 'r' of the reversible adiabatic compression is variable. The ratio of the specific heats is γ.*

Show that if the work done in the cycle is a maximum then,

$$(\gamma - 1) \log_e \frac{x}{r} + \frac{1}{r^{\gamma-1}} - 1 = 0.$$

Solution. Refer Fig. 3.1.

$$\frac{V_3}{V_1} = x \; ; \; \frac{V_4}{V_1} = r$$

During isotherms, since compression ratio = expansion ratio

$$\therefore \qquad \frac{V_3}{V_4} = \frac{V_2}{V_1}$$

Also $$\frac{V_3}{V_4} = \frac{V_3}{V_1} \times \frac{V_1}{V_4} = x \times \frac{1}{r} = \frac{x}{r}$$

Work done per kg of the gas

$$= \text{Heat supplied} - \text{Heat rejected} = RT_1 \log_e \frac{x}{r} - RT_2 \log_e \frac{x}{r}$$

$$= R(T_1 - T_2) \log_e \frac{x}{r} = RT_2 \left(\frac{T_1}{T_2} - 1\right) \log_e \frac{x}{r}$$

But $$\frac{T_1}{T_2} = \left(\frac{V_4}{V_1}\right)^{\gamma-1} = (r)^{\gamma-1}$$

$\therefore$ Work done per kg of the gas,

$$W = RT_2\, (r^{\gamma-1} - 1) \log_e \frac{x}{r}$$

Differentiating W w.r.t. 'r' and equating to zero

$$\frac{dW}{dr} = RT_2 \left[(r^{\gamma-1} - 1)\left\{\frac{r}{x} \times (-xr^{-2})\right\} + \log_e \frac{x}{r}\{(\gamma-1)r^{\gamma-2}\}\right] = 0$$

or $$(r^{\gamma-1} - 1)\left(-\frac{1}{r}\right) + (\gamma - 1) \times r^{\gamma-2} \log_e \frac{x}{r} = 0$$

or $$-r^{\gamma-2} + \frac{1}{r} + r^{\gamma-2}\,(\gamma - 1) \log_e \frac{x}{r} = 0$$

or $$r^{\gamma-2}\left\{-1 + \frac{1}{r \cdot r^{\gamma-2}} + (\gamma-1)\log_e \frac{x}{r}\right\} = 0$$

or $$-1 + \frac{1}{r \cdot r^{\gamma-2}} + (\gamma - 1) \log_e \frac{x}{r} = 0$$

$$(\gamma - 1) \log_e \frac{x}{r} + \frac{1}{r^{\gamma-1}} - 1 = 0. \quad \textbf{Proved.}$$

3.4. CONSTANT VOLUME OR OTTO CYCLE

This cycle is so named as it was conceived by 'Otto'. On this cycle, petrol, gas and many types of oil engines work. *It is the standard of comparison for internal combustion engines.*

Fig. 3.5 (*a*) and (*b*) shows the theoretical *p-V* diagram and *T-s* diagrams of this cycle respectively.

The point 1 represents that cylinder is full of air with volume V_1, pressure p_1 and absolute temperature T_1.

Line 1-2 represents the adiabatic compression of air due to which p_1, V_1 and T_1 change to p_2, V_2 and T_2 respectively.

Line 2-3 shows the supply of heat to the air at constant volume so that p_2 and T_2 change to p_3 and T_3 (V_3 being the same as V_2).

Line 3-4 represents the adiabatic expansion of the air. During expansion p_3, V_3 and T_3 change to a final value of p_4, V_4 or V_1 and T_4 respectively.

Line 4-1 shows the rejection of heat by air at constant volume till original state (point 1) reaches.

Consider *1 kg of air* (working substance) :

Heat supplied at constant volume $= c_v(T_3 - T_2)$.

Heat rejected at constant volume $= c_v(T_4 - T_1)$.

But, work done = heat supplied – heat rejected

$$= c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\therefore \quad \text{Efficiency} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_3 - T_2) - c_v(T_4 - T_1)}{c_v(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2} \quad \ldots(i)$$

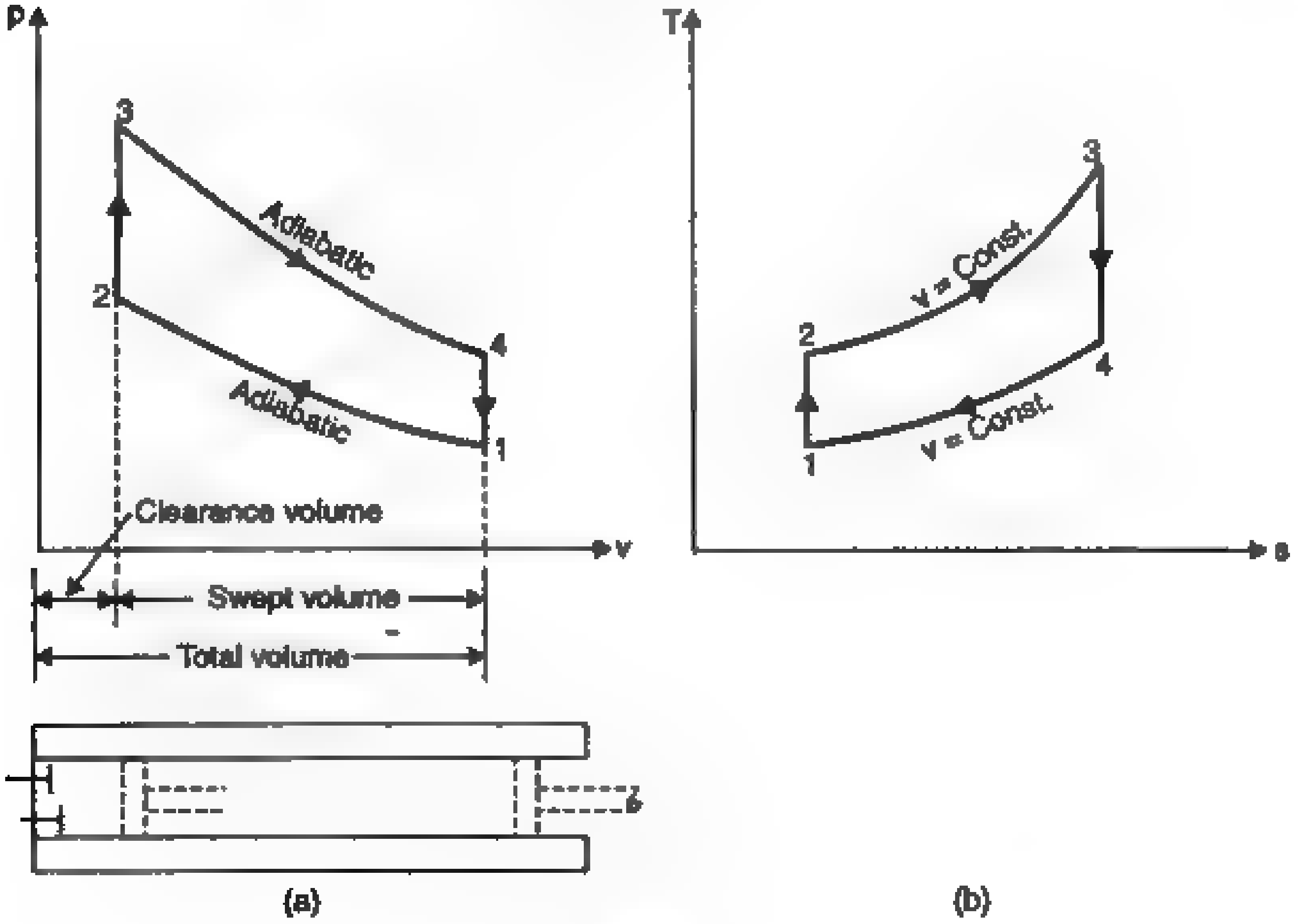


Fig. 3.5

Let compression ratio, $r_c\ (= r) = \dfrac{v_1}{v_2}$

and expansion ratio, $r_e\ (= r) = \dfrac{v_4}{v_3}$

(These *two ratios are same* in this cycle)

As $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$

Then, $$T_2 = T_1 \,.\, (r)^{\gamma-1}$$

Similarly, $$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1}$$

or $$T_3 = T_4 \,.\, (r)^{\gamma-1}$$

Inserting the values of T_2 and T_3 in equation (*i*), we get

$$\eta_{otto} = 1 - \frac{T_4 - T_1}{T_4 \,.(r)^{\gamma-1} - T_1 \,.(r)^{\gamma-1}} = 1 - \frac{T_4 - T_1}{r^{\gamma-1}(T_4 - T_1)}$$

$$= 1 - \frac{1}{(r)^{\gamma-1}} \qquad ...(3.3)$$

This expression is known as the air standard efficiency of the Otto cycle.

It is clear from the above expression that efficiency increases with the increase in the value of r, which means we can have maximum efficiency by increasing r to a considerable extent, but *due to practical difficulties its value is limited to about 9.*

The *net work done per kg* in the Otto cycle can also be expressed in terms of p, v. If p is expressed in bar *i.e.* 10^5 N/m², then work done

$$W = \left(\frac{p_3v_3 - p_4v_4}{\gamma-1} - \frac{p_2v_2 - p_1v_1}{\gamma-1}\right) \times 10^2 \text{ kJ} \qquad ...(3.4)$$

Also $$\frac{p_3}{p_4} = r^{\gamma} = \frac{p_2}{p_1}$$

$\therefore$ $$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p$$

where r_p stands for *pressure ratio.*

and $$v_1 = rv_2 = v_4 = rv_3 \qquad \left[\because\ \frac{v_1}{v_2} = \frac{v_4}{v_3} = r\right]$$

$\therefore$ $$W = \frac{1}{\gamma-1}\left[p_4v_4\left(\frac{p_3v_3}{p_4v_4} - 1\right) - p_1v_1\left(\frac{p_2v_2}{p_1v_1} - 1\right)\right]$$

$$= \frac{1}{\gamma-1}\left[p_4v_4\left(\frac{p_3}{p_4r} - 1\right) - p_1v_1\left(\frac{p_2}{p_1r} - 1\right)\right]$$

$$= \frac{v_1}{\gamma-1}\left[p_4\,(r^{\gamma-1} - 1) - p_1\,(r^{\gamma-1} - 1)\right]$$

$$= \frac{v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(p_4 - p_1)\right]$$

$$= \frac{p_1 v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(r_p - 1)\right] \quad \text{...[3.4 (a)]}$$

Mean effective pressure (p_m) is given by :

$$p_m = \left[\left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}\right) \div (v_1 - v_2)\right] \text{ bar} \quad \text{...(3.5)}$$

Also
$$p_m = \frac{\left[\frac{p_1 v_1}{\gamma - 1}(r^{\gamma-1} - 1)(r_p - 1)\right]}{(v_1 - v_2)}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)]}{v_1 - \frac{v_1}{r}}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)]}{v_1\left(\frac{r-1}{r}\right)}$$

i.e.,
$$\mathbf{p_m} = \frac{p_1 r[(r^{\gamma-1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)} \quad \text{...(3.6)}$$

Example 3.7. *The efficiency of an Otto cycle is 60% and $\gamma = 1.5$. What is the compression ratio ?*

Solution. Efficiency of Otto cycle, $\eta = 60\%$

Ratio of specific heats, $\gamma = 1.5$

Compression ratio, $r = ?$

Efficiency of Otto cycle is given by

$$\eta_{Otto} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

$$0.6 = 1 - \frac{1}{(r)^{1.5 - 1}}$$

or
$$\frac{1}{(r)^{0.5}} = 0.4 \quad \text{or} \quad (r)^{0.5} = \frac{1}{0.4} = 2.5 \quad \text{or} \quad r = 6.25$$

Hence, *compression ratio* **= 6.25. (Ans.)**

Example 3.8. *An engine of 250 mm bore and 375 mm stroke works on Otto cycle. The clearance volume is 0.00263 m³. The initial pressure and temperature are 1 bar and 50°C. If the maximum pressure is limited to 25 bar, find the following :*

(i) The air standard efficiency of the cycle.

(ii) The mean effective pressure for the cycle.

Assume the ideal conditions.

Solution. Bore of the engine, $D = 250$ mm $= 0.25$ m
Stroke of the engine, $L = 375$ mm $= 0.375$ m
Clearance volume, $V_c = 0.00263$ m^3
Initial pressure, $p_1 = 1$ bar
Initial temperature, $T_1 = 50 + 273 = 323$ K

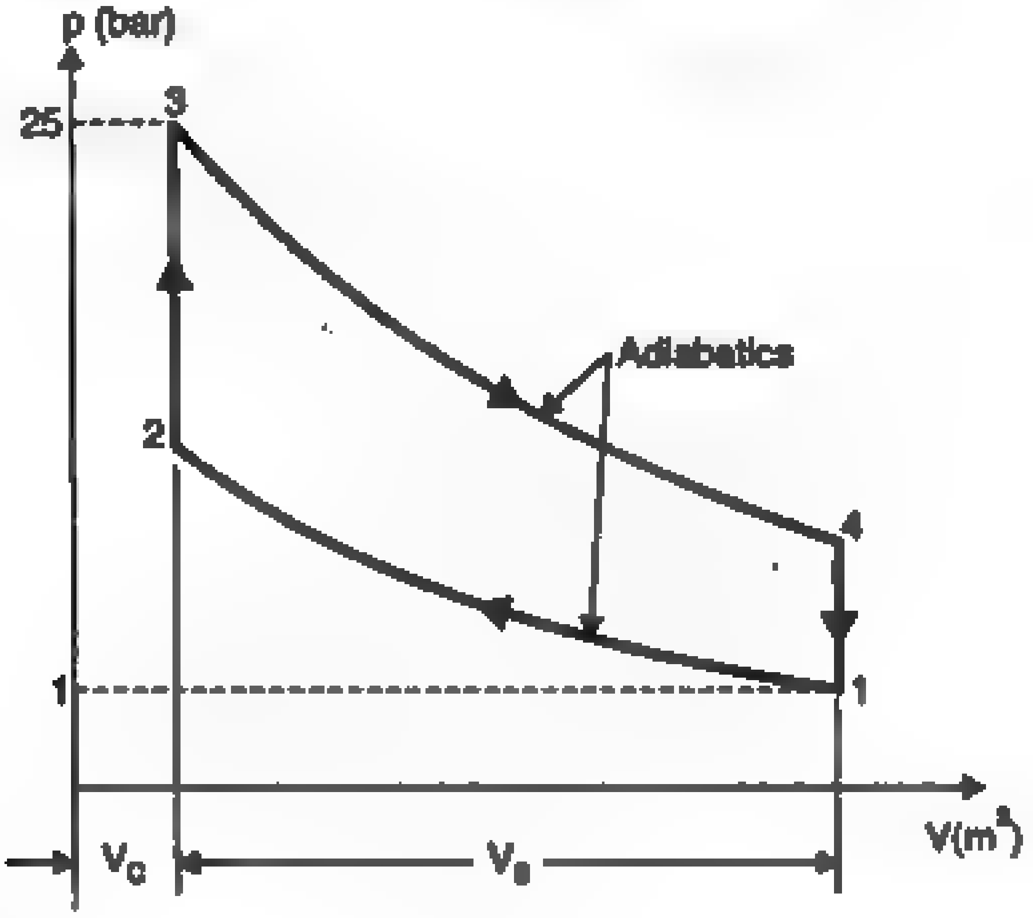


Fig. 3.6

Maximum pressure, $p_3 = 25$ bar
Swept volume, $V_s = \pi/4\ D^2L = \pi/4 \times 0.25^2 \times 0.375 = 0.0184$ m^3

Compression ratio, $r = \dfrac{V_s + V_c}{V_c} = \dfrac{0.0184 + 0.00263}{0.00263} = 8.$

(i) Air standard efficiency :

The air standard efficiency of Otto cycle is given by

$$\eta_{Otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(8)^{1.4-1}} = 1 - \frac{1}{(8)^{0.4}}$$

$$= 1 - 0.435 = \mathbf{0.565} \quad \text{or} \quad \mathbf{56.5\%.} \quad \text{(Ans.)}$$

(ii) Mean effective pressure, p_m :

For adiabatic (or isentropic) process 1-2

$$p_1V_1^\gamma = p_2V_2^\gamma$$

or $$p_2 = p_1\left(\frac{V_1}{V_2}\right)^\gamma = 1 \times (r)^{1.4} = 1 \times (8)^{1.4} = 18.38 \text{ bar}$$

$\therefore$ Pressure ratio, $r_p = \dfrac{p_3}{p_2} = \dfrac{25}{18.38} = 1.36$

The mean effective pressure is given by

$$p_m = \frac{p_1r[(r^{\gamma-1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)} = \frac{1 \times 8[\{(8)^{1.4-1} - 1\}(1.36 - 1)]}{(1.4 - 1)(8 - 1)} \qquad \text{...[Eqn. (3.6)]}$$

$$= \frac{8(2.297-1)(0.36)}{0.4 \times 7} = 1.334 \text{ bar}$$

Hence *mean effective pressure* = **1.334 bar. (Ans.)**

Example 3.9. *The minimum pressure and temperature in an Otto cycle are 100 kPa and 27°C. The amount of heat added to the air per cycle is 1500 kJ/kg.*

(i) Determine the pressures and temperatures at all points of the air standard Otto cycle.

(ii) Also calculate the specific work and thermal efficiency of the cycle for a compression ratio of 8 : 1.

Take for air : c_v = 0.72 kJ/kg K, and γ = 1.4. **(GATE, 1998)**

Solution. Refer Fig. 3.7. *Given :* p_1 = 100 kPa = 10^5 N/m² or 1 bar ;

T_1 = 27 + 273 = 300 K ; Heat added = 1500 kJ/kg ;

r = 8 : 1 ; c_v = 0.72 kJ/kg ; γ = 1.4.

Consider 1 kg of air.

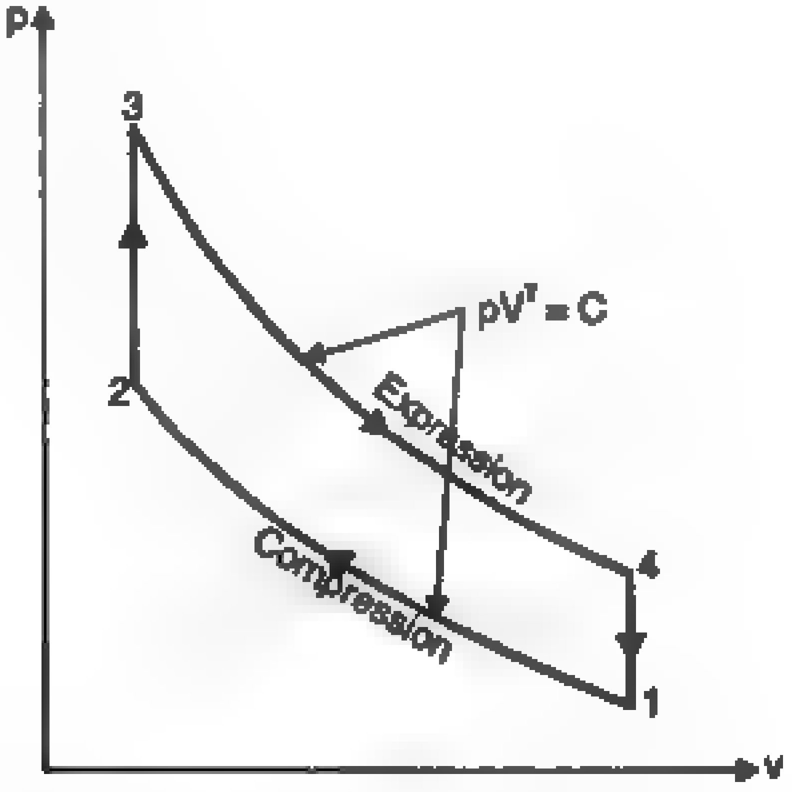


Fig. 3.7

(i) Pressures and temperatures at all points :

Adiabatic Compression process 1-2 :

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (8)^{1.4-1} = 2.297$$

$\therefore$ $T_2 = 300 \times 2.297 =$ **689.1 K. (Ans.)**

Also $p_1 v_1^{\gamma} = p_2 v_2^{\gamma}$

or $$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (8)^{1.4} = 18.379$$

$\therefore$ $p_2 = 1 \times 18.379 =$ **18.379 bar. (Ans.)**

Constant volume process 2-3 :

Heat added during the process,

$$c_v(T_3 - T_2) = 1500$$

or $$0.72\,(T_3 - 689.1) = 1500$$

or $$T_3 = \frac{1500}{0.72} + 689.1 = 2772.4 \text{ K. (Ans.)}$$

Also, $$\frac{p_2}{T_2} = \frac{p_3}{T_3} \Rightarrow p_3 = \frac{p_2 T_3}{T_2} = \frac{18.379 \times 2772.4}{689.1} = 73.94 \text{ bar. (Ans.)}$$

Adiabatic Expansion process 3-4 :

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = (r)^{\gamma-1} = (8)^{1.4-1} = 2.297$$

$\therefore$ $$T_4 = \frac{T_3}{2.279} = \frac{2772.4}{2.297} = 1206.9 \text{ K. (Ans.)}$$

Also, $$p_3 v_3^{\gamma} = p_4 v_4^{\gamma} \Rightarrow p_4 = p_3 \times \left(\frac{v_3}{v_4}\right)^{\gamma} = 73.94 \times \left(\frac{1}{8}\right)^{1.4} = 4.023 \text{ bar. (Ans.)}$$

(*ii*) Specific work and thermal efficiency :

Specfic work = Heat added – Heat rejected

$$= c_v\,(T_3 - T_2) - c_v(T_4 - T_1) = c_v\,[(T_3 - T_2) - (T_4 - T_1)]$$
$$= 0.72\,[(2772.4 - 689.1) - (1206.9 - 300)] = \mathbf{847\ kJ/kg.}\ \text{(Ans.)}$$

Thermal efficiency, $$\eta_{th} = 1 - \frac{1}{(r)^{\gamma-1}}$$
$$= 1 - \frac{1}{(8)^{1.4-1}} = 0.5647 \text{ or } \mathbf{56.47\%.}\ \text{(Ans.)}$$

Example 3.10. *An air standard Otto cycle has a volumetric compression ratio of 6, the lowest cycle pressure of 0.1 MPa and operates between temperature limits of 27°C and 1569°C.*

(i) Calculate the temperature and pressure after the isentropic expansion (ratio of specific heats = 1.4).

(ii) Since it is observed that values in (i) are well above the lowest cycle operating conditions, the expansion process was allowed to continue down to a pressure of 0.1 MPa. Which process is required to complete the cycle ? Name the cycle so obtained.

(iii) Determine by what percentage the cycle efficiency has been improved. **(GATE, 1994)**

Solution. Refer Fig. 3.8. *Given :* $\frac{v_1}{v_2} = \frac{v_4}{v_3} = r = 6$; $p_1 = 0.1$ MPa $= 1$ bar ; $T_1 = 27 + 273 = 300$ K ; $T_3 = 1569 + 273 = 1842$ K ; $\gamma = 1.4$.

(*i*) Temperature and pressure after the isentropic expansion, T_4, p_4 :

Consider 1 kg of air :

For *the compression process 1-2 :*

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \Rightarrow p_2 = p_1 \times \left(\frac{v_1}{v_2}\right)^{\gamma} = 1 \times (6)^{1.4} = 12.3 \text{ bar}$$

Also $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (6)^{1.4-1} = 2.048$$

$\therefore$ $$T_2 = 300 \times 2.048 = 614.4 \text{ K}$$

For the constant volume process 2-3 :

$$\frac{p_2}{T_2} = \frac{p_3}{T_3} \Rightarrow p_3 = \frac{p_2 T_3}{T_2} = 12.3 \times \frac{1842}{614.4} = 36.9 \text{ bar}$$

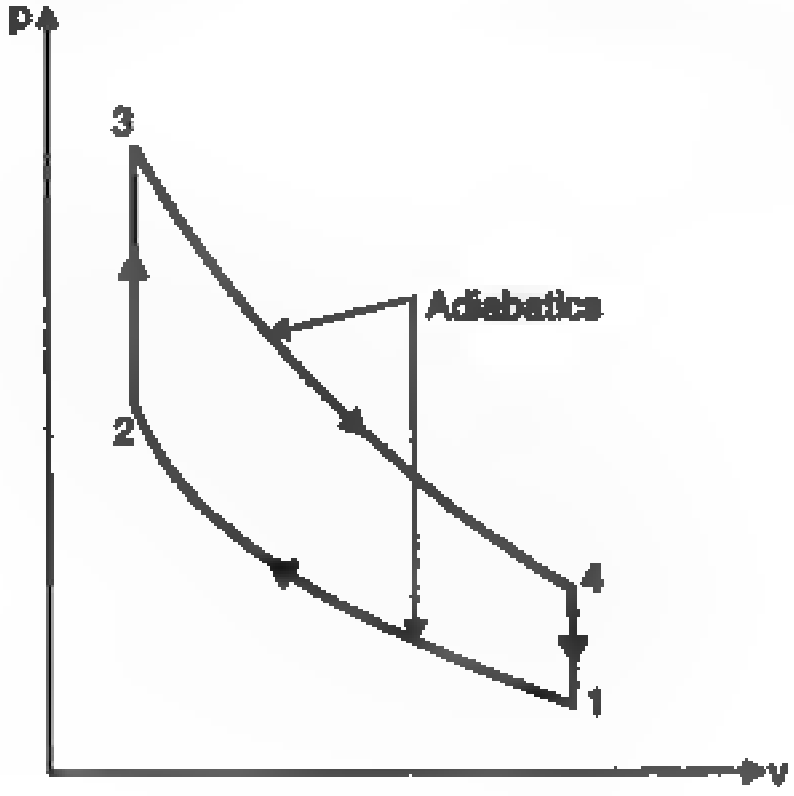


Fig. 3.8

For the expansion process 3-4 :

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = (6)^{1.4-1} = 2.048$$

$\therefore$
$$T_4 = \frac{T_3}{2.048} = \frac{1842}{2.048} = \mathbf{900\ K.} \quad \textbf{(Ans.)}$$

Also
$$p_3 v_3^{\gamma} = p_4 v_4^{\gamma} \Rightarrow p_4 = p_3 \times \left(\frac{v_3}{v_4}\right)^{\gamma}$$

or
$$p_4 = 36.9 \times \left(\frac{1}{6}\right)^{1.4} = \mathbf{3\ bar.} \quad \textbf{(Ans.)}$$

(*ii*) **Process required to complete the cycle :**

Process required to complete the cycle is the ***constant pressure*** **scavenging.**

The cycle is called Atkinson cycle (Refer Fig. 3.9).

(*iii*) **Percentage improvement/increase in efficiency :**

$$\eta_{Otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(6)^{1.4-1}} = 0.5116 \quad \text{or} \quad \mathbf{51.16\%.} \quad \textbf{(Ans.)}$$

$$\eta_{Atkinson} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}}$$

$$= \frac{c_v(T_3 - T_2) - c_p(T_5 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{c_p(T_5 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{\gamma(T_5 - T_1)}{(T_3 - T_2)}$$

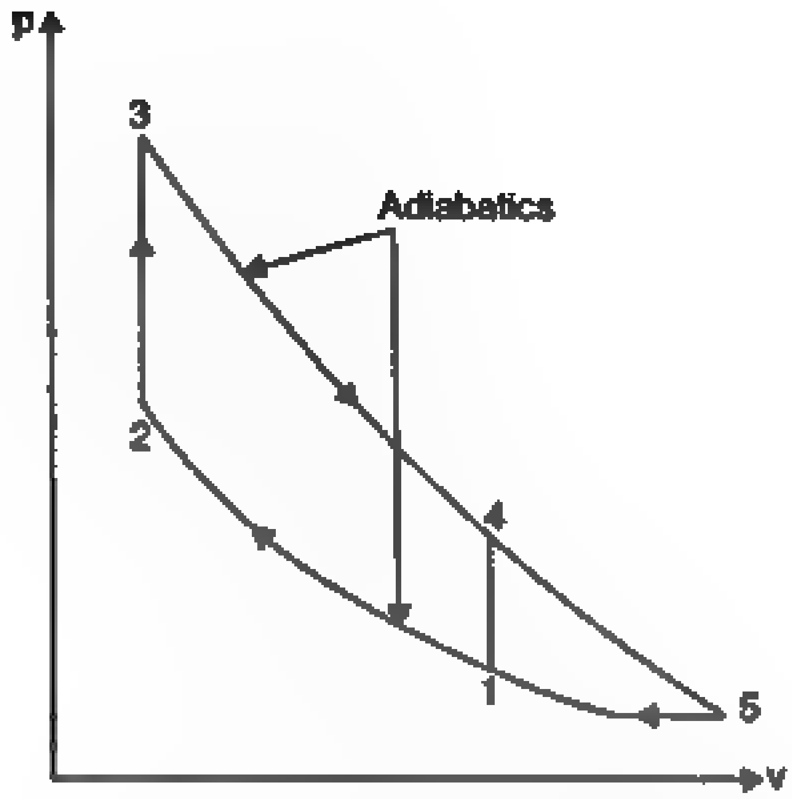


Fig. 3.9. Atkinson cycle.

Now, $$\frac{T_5}{T_3} = \left(\frac{p_5}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{or} \quad T_5 = 1842 \times \left(\frac{1.0}{36.9}\right)^{\frac{1.4-1}{1.4}} = 657 \text{ K}$$

$\therefore$ $$\eta_{\text{Atkinson}} = 1 - \frac{1.4(657 - 300)}{(1842 - 614.4)} = 0.5929 \quad \text{or} \quad \mathbf{59.29\%.}$$

$\therefore$ **Improvement in efficiency = 59.29 – 51.16 = 8.13%. (Ans.)**

Example 3.11. *A certain quantity of air at a pressure of 1 bar and temperature of 70°C is compressed adiabatically until the pressure is 7 bar in Otto cycle engine. 465 kJ of heat per kg of air is now added at constant volume. Determine :*

(i) Compression ratio of the engine.

(ii) Temperature at the end of compression.

(iii) Temperature at the end of heat addition.

Take for air c_p = 1.0 kJ/kg K, c_v = 0.706 kJ/kg K.

Show each operation on p-V and T-s diagrams.

Solution. Refer Fig 3.10.

Initial pressure, p_1 = 1 bar

Initial temperature, T_1 = 70 + 273 = 343 K

Pressure after adiabatic compression, p_2 = 7 bar

Heat addition at constant volume, Q_s = 465 kJ/kg of air

Specific heat at constant pressure, c_p = 1.0 kJ/kg K

Specific heat at constant volume, c_v = 0.706 kJ/kg K

$\therefore$ $$\gamma = \frac{c_p}{c_v} = \frac{1.0}{0.706} = 1.41$$

(i) Compression ratio of engine, r :

According to *adiabatic compression* 1-2

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

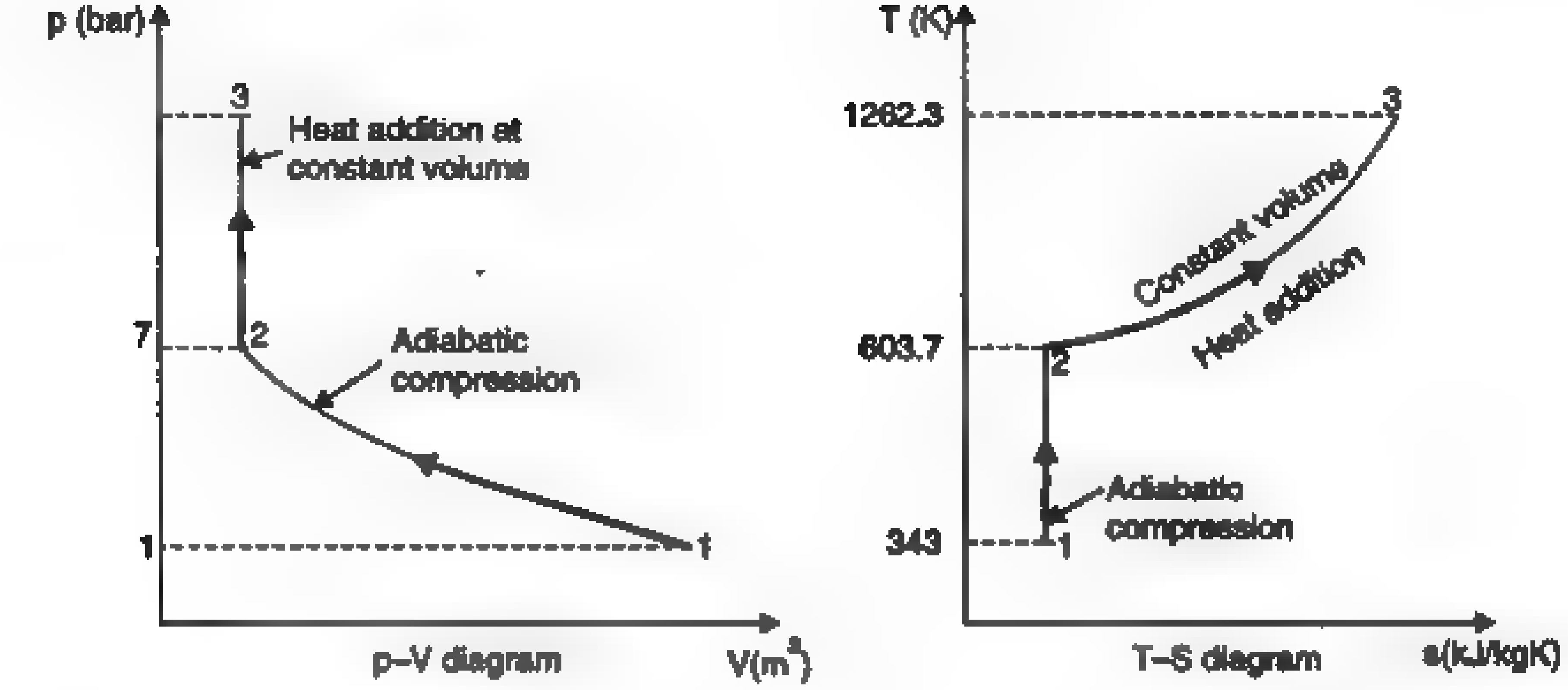


Fig. 3.10

or $$\left(\frac{V_1}{V_2}\right)^{\gamma} = \frac{p_2}{p_1}$$

or $$(r)^{\gamma} = \frac{p_2}{p_1} \qquad \left(\because \frac{V_1}{V_2} = r\right)$$

or $$r = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = \left(\frac{7}{1}\right)^{\frac{1}{1.41}} = (7)^{0.709} = 3.97$$

Hence *compression ratio of the engine* = **3.97. (Ans.)**

(*ii*) **Temperature at the end of compression, T_2 :**

In case of *adiabatic compression 1-2*,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = (3.97)^{1.41 - 1} = 1.76$$

$\therefore$ $$T_2 = 1.76\ T_1 = 1.76 \times 343 = 603.7 \text{ K or } 330.7°\text{C}$$

Hence *temperature at the end of compression* = **330.7°C. (Ans.)**

(*iii*) **Temperature at the end of heat addition, T_3 :**

According to *constant volume heating operation 2-3*

$$Q_s = c_v (T_3 - T_2) = 465$$

$$0.706 (T_3 - 603.7) = 465$$

or $$T_3 - 603.7 = \frac{\text{[redacted]}}{0.706}$$

or $$T_3 = \frac{465}{0.706} + 603.7 = 1262.3 \text{ K or } 989.3°\text{C}$$

Hence *temperature at the end of heat addition* = **989.3°C. (Ans.)**

Example 3.12. *In a constant volume 'Otto cycle', the pressure at the end of compression is 15 times that at the start, the temperature of air at the beginning of compression is 38°C and maximum temperature attained in the cycle is 1950°C. Determine :*

(i) Compression ratio.
(ii) Thermal efficiency of the cycle.
(iii) Work done.
Take γ for air = 1.4.

Solution. Refer Fig. 3.11.

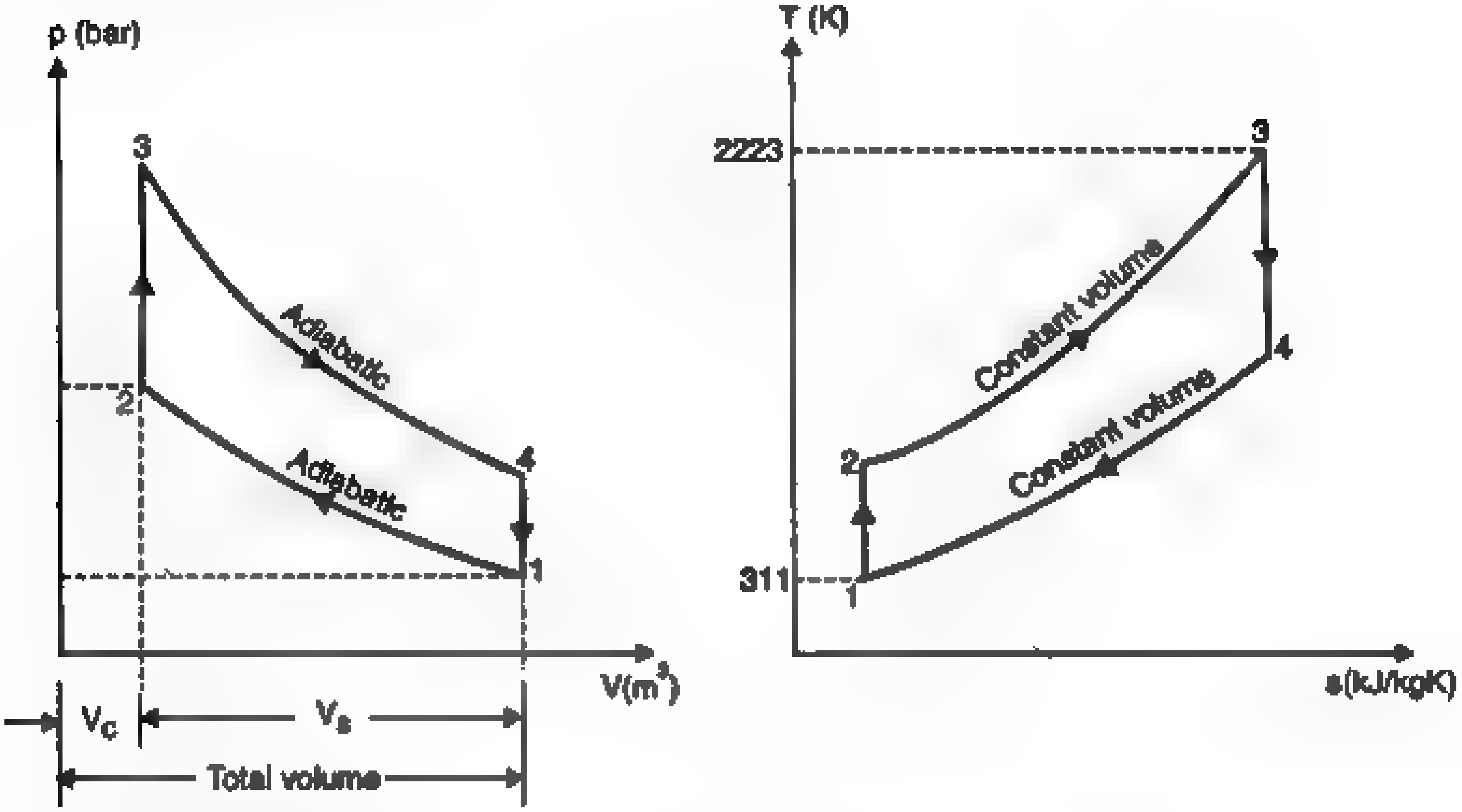


Fig. 3.11

Initial temperature, $T_1 = 38 + 273 = 311$ K

Maximum temperature, $T_3 = 1950 + 273 = 2223$ K.

(i) **Compression ratio, r :**

For *adiabatic compression 1-2,*

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

or $$\left(\frac{V_1}{V_2}\right)^{\gamma} = \frac{p_2}{p_1}$$

But $$\frac{p_2}{p_1} = 15 \qquad \text{...(Given)}$$

$\therefore$ $$(r)^{\gamma} = 15 \qquad \left[\because \; r = \frac{V_1}{V_2}\right]$$

or $$(r)^{1.4} = 15$$

or $$r = (15)^{\frac{1}{1.4}} = (15)^{0.714} = 6.9$$

Hence *compression ratio* = **6.9. (Ans.)**

(ii) **Thermal efficiency :**

Thermal efficiency, $$\eta_{th} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(6.9)^{1.4-1}} = \textbf{0.538 or 53.8\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Work done :**

Again, for *adiabatic compression 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (6.9)^{1.4-1} = (6.9)^{0.4} = 2.16$$

or $$T_2 = T_1 \times 2.16 = 311 \times 2.16 = 671.7 \text{ K or } 398.7°\text{C}$$

For *adiabatic expansion process 3-4,*

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = (r)^{\gamma-1} = (6.9)^{0.4} = 2.16$$

or $$T_4 = \frac{T_3}{2.16} = \frac{\text{[redacted]}}{2.16} = 1029 \text{ K or } 756°\text{C}$$

Heat supplied *per kg of air*

$$= c_v(T_3 - T_2) = 0.717(2223 - 671.7)$$

$$= 1112.3 \text{ kJ/kg of air} \qquad \left[c_v = \frac{R}{\gamma-1} = \frac{0.287}{1.4-1} = 0.717 \text{ kJ/ kg K}\right]$$

Heat rejected per kg of air

$$= c_v(T_4 - T_1) = 0.717(1029 - 311)$$

$$= 514.8 \text{ kJ/kg of air}$$

∴ *Work done* $= \text{Heat supplied} - \text{Heat rejected}$

$$= 1112.3 - 514.8$$

$$= \mathbf{597.5 \text{ kJ or } 597500 \text{ N-m.}} \quad \textbf{(Ans.)}$$

☞**Example 3.13.** *An engine working on Otto cycle has a volume of 0.45 m^3, pressure 1 bar and temperature 30°C at the beginning of compression stroke. At the end of compression stroke, the pressure is 11 bar. 210 kJ of heat is added at constant volume. Determine :*

(*i*) *Pressures, temperatures and volumes at salient points in the cycle.*

(*ii*) *Percentage clearance.*

(*iii*) *Efficiency.*

(*iv*) *Net work per cycle.*

(*v*) *Mean effective pressure.*

(*vi*) *Ideal power developed by the engine if the number of working cycles per minute is 210.*

Assume the cycle is reversible.

Solution. Refer Fig. 3.12

Volume, $V_1 = 0.45$ m^3

Initial pressure, $p_1 = 1$ bar

Initial temperature, $T_1 = 30 + 273 = 303$ K

Pressure at the end of compression stroke, $p_2 = 11$ bar

Heat added at constant volume = 210 kJ

Number of working cycles/min. = 210.

(*i*) **Pressures, temperatures and volumes at salient points :**

For *adiabatic compression 1-2,*

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

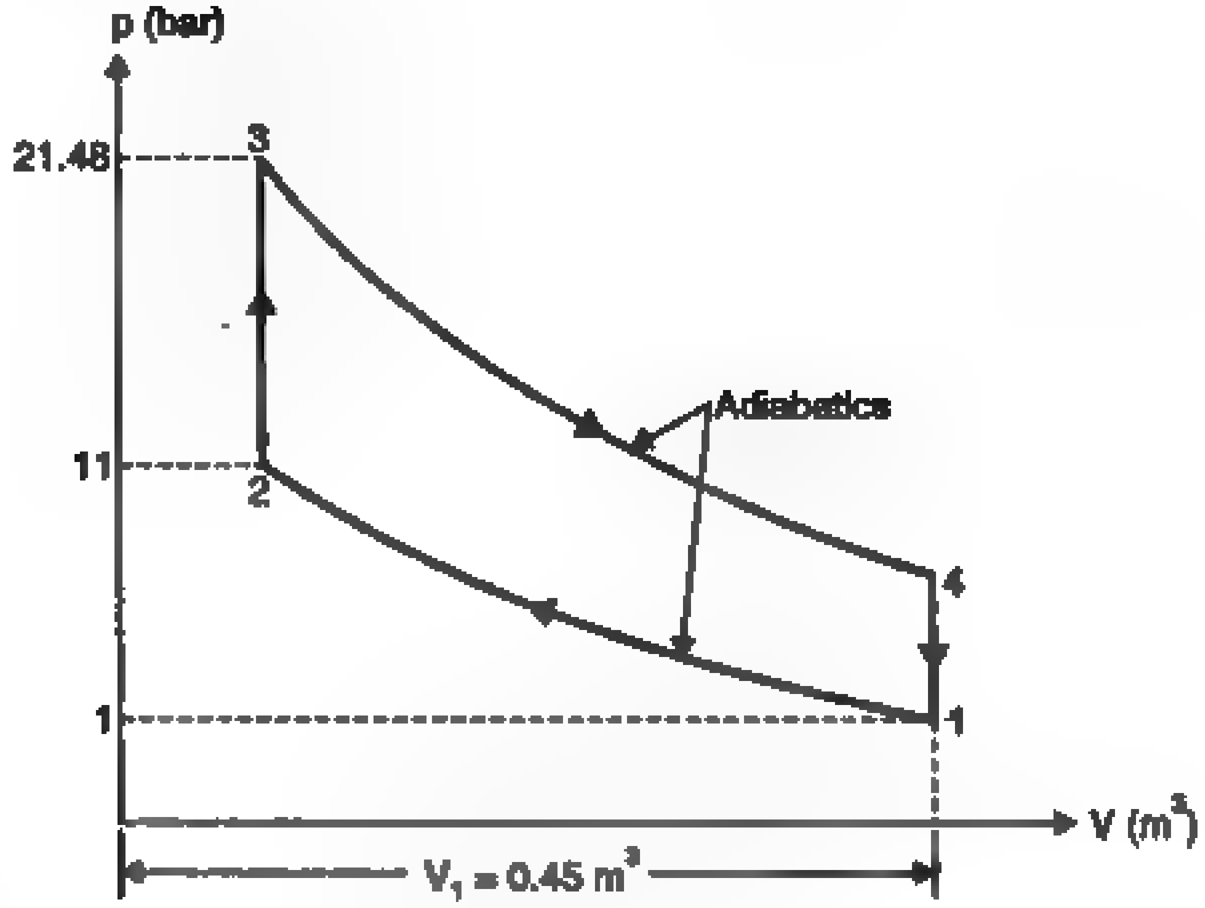


Fig. 3.12

or $$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = (r)^{\gamma} \quad \text{or} \quad r = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = \left(\frac{11}{1}\right)^{\frac{1}{1.4}} = (11)^{0.714} = 5.5$$

Also $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (5.5)^{1.4-1} = 1.977 \approx 1.98$$

$\therefore$ $$T_2 = T_1 \times 1.98 = 303 \times 1.98 = 600 \text{ K.} \quad \textbf{(Ans.)}$$

Applying gas laws to points 1 and 2, we have

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$\therefore$ $$V_2 = \frac{T_2}{T_1} \times \frac{p_1}{p_2} \times V_1 = \frac{600 \times 1 \times 0.45}{303 \times 11} = 0.081 \text{ m}^3. \quad \textbf{(Ans.)}$$

The heat supplied during the *process 2-3* is given by :

$$Q_s = m\, c_v (T_3 - T_2)$$

where $$m = \frac{p_1V_1}{RT_1} = \frac{1 \times 10^5 \times 0.45}{287 \times 303} = 0.517 \text{ kg}$$

$\therefore$ $$210 = 0.517 \times 0.71 (T_3 - 600)$$

or $$T_3 = \frac{[redacted]}{0.517 \times 0.71} + 600 = 1172 \text{ K.} \quad \textbf{(Ans.)}$$

For the *constant volume process 2-3*,

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

$\therefore$ $$p_3 = \frac{T_3}{T_2} \times p_2 = \frac{1172}{600} \times 11 = 21.48 \text{ bar.} \quad \textbf{(Ans.)}$$

$$V_3 = V_2 = 0.081 \text{ m}^3. \quad \textbf{(Ans.)}$$

For the *adiabatic (or isentropic) process 3-4,*

$$p_3V_3^{\gamma} = p_4V_4^{\gamma}$$

$$p_4 = p_3 \times \left(\frac{V_3}{V_4}\right)^{\gamma} = p_3 \times \left(\frac{1}{r}\right)^{\gamma}$$

$$= 21.48 \times \left(\frac{1}{\text{[illegible]}}\right)^{1.4} = \mathbf{1.97\ bar.} \quad \textbf{(Ans.)}$$

Also $$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} = \left(\frac{1}{5.5}\right)^{1.4-1} = 0.505$$

$\therefore$ $$T_4 = 0.505\ T_3 = 0.505 \times 1172 = \mathbf{591.8\ K.} \quad \textbf{(Ans.)}$$

$$V_4 = V_1 = \mathbf{0.45\ m^3.} \quad \textbf{(Ans.)}$$

(*ii*) **Percentage clearance :**

Percentage clearance

$$= \frac{V_c}{V_s} = \frac{V_2}{V_1 - V_2} \times 100 = \frac{0.081}{0.45 - 0.081} \times 100$$

$$= \mathbf{21.95\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Efficiency :**

The heat rejected per cycle is given by

$$Q_r = mc_v(T_4 - T_1)$$

$$= 0.517 \times 0.71\ (591.8 - 303) = 106\ \text{kJ}$$

The air-standard efficiency of the cycle is given by

$$\eta_{otto} = \frac{Q_s - Q_r}{Q_s} = \frac{210 - 106}{210} = \mathbf{0.495\ or\ 49.5\%.} \quad \textbf{(Ans.)}$$

[Alternatively :

$$\eta_{otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(5.5)^{1.4-1}} = \mathbf{0.495\ or\ 49.5\%.} \quad \textbf{(Ans.)}$$]

(*iv*) **Mean effective pressure, p_m :**

The mean effective pressure is given by

$$p_m = \frac{W\ \text{(work done)}}{V_s\text{(swept volume)}} = \frac{Q_s - Q_r}{(V_1 - V_2)}$$

$$= \frac{(210 - 106) \times 10^3}{(0.45 - 0.081) \times 10^5} = \mathbf{2.818\ bar.} \quad \textbf{(Ans.)}$$

(*v*) **Power developed, P :**

Power developed, P = Work done per second

= Work done per cycle × Number of cycles per second

= (210 – 106) × (210/60) = **364 kW. (Ans.)**

Example 3.14. (*a*) *Show that the compression ratio for the maximum work to be done per kg of air in an Otto cycle between upper and lower limits of absolute temperatures T_3 and T_1 is given by*

$$r = \left(\frac{T_3}{T_1}\right)^{1/2(\gamma-1)}$$

(b) Determine the air-standard efficiency of the cycle when the cycle develops maximum work with the temperature limits of 310 K and 1220 K and working fluid is air. What will be the percentage change in efficiency if helium is used as working fluid instead of air ? The cycle operates between the same temperature limits for maximum work development.

Consider that all conditions are ideal.

Solution. Refer Fig. 3.13.

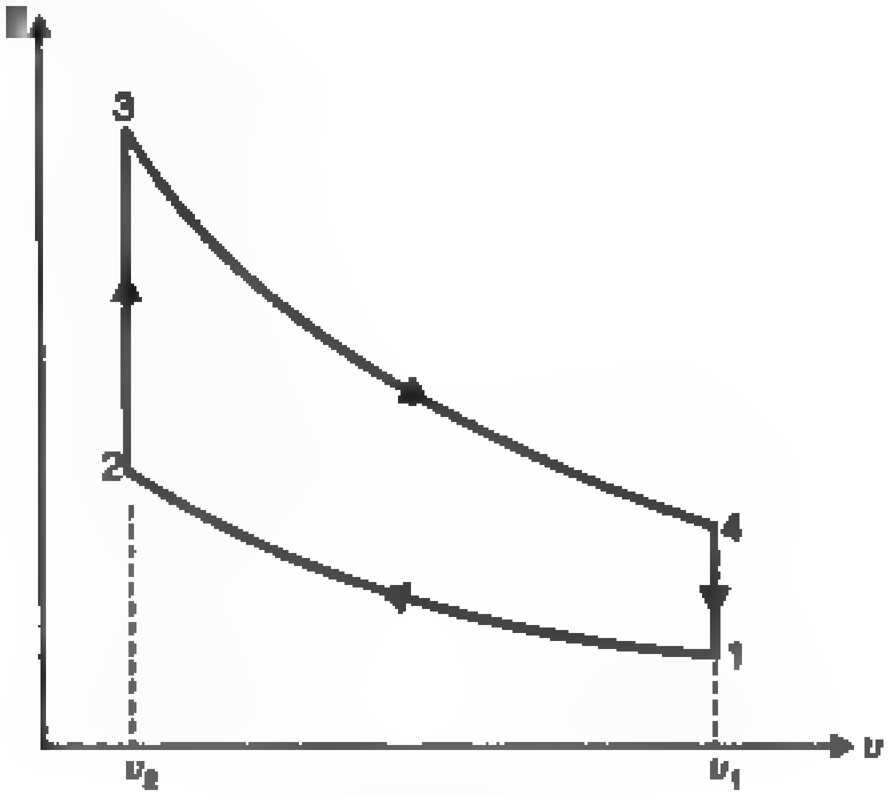


Fig. 3.13

(*a*) The work done per kg of fluid in the cycle is given by

$$W = Q_s - Q_r = c_v (T_3 - T_2) - c_v (T_4 - T_1)$$

But $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = (r)^{\gamma - 1}$$

$\therefore$ $$T_2 = T_1 \,.\, (r)^{\gamma - 1} \quad ...(i)$$

Similarly, $$T_3 = T_4 \,.\, (r)^{\gamma - 1} \quad ...(ii)$$

$\therefore$ $$W = c_v \left[T_3 - T_1 \,.\, (r)^{\gamma - 1} - \frac{T_3}{(r)^{\gamma - 1}} + T_1\right] \quad ...(iii)$$

This expression is a function of r when T_3 and T_1 are fixed. The value of W will be maximum when

$$\frac{dW}{dr} = 0.$$

$\therefore$ $$\frac{dW}{dr} = - T_1 \,.\, (\gamma - 1)\ (r)^{\gamma - 2} - T_3 (1 - \gamma)\ (r)^{-\gamma} = 0$$

or $$T_3 (r)^{-\gamma} = T_1 (r)^{\gamma - 2}$$

or $$\frac{T_3}{T_1} = (r)^{2(\gamma - 1)}$$

$\therefore$ $$r = \left(\frac{T_3}{T_1}\right)^{1/2(\gamma - 1)}$$ **Proved.**

(*b*) **Change in efficiency :**

For *air* $\gamma = 1.4$

$$\therefore \qquad r = \left(\frac{T_3}{T_1}\right)^{1/2(1.4-1)} = \left(\frac{1220}{310}\right)^{1/0.8} = 5.54$$

The air-standard efficiency is given by

$$\eta_{otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(5.54)^{1.4-1}} = 0.495 \text{ or } 49.5\%. \quad \textbf{(Ans.)}$$

If *helium* is used, then the values of

$$c_p = 5.22 \text{ kJ/kgK} \quad \text{and} \quad c_v = 3.13 \text{ kJ/kg K}$$

$$\therefore \qquad \gamma = \frac{c_p}{c_v} = \frac{5.22}{3.13} = 1.67$$

The compression ratio for maximum work for the temperature limits T_1 and T_3 is given by

$$r = \left(\frac{T_3}{T_1}\right)^{1/2(\gamma-1)} = \left(\frac{1220}{310}\right)^{1/2(1.67-1)} = 2.77$$

The air-standard efficiency is given by

$$\eta_{otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(2.77)^{1.67-1}} = 0.495 \quad \text{or} \quad 49.5\%.$$

Hence change in efficiency is nil. (Ans.)

Example 3.15. (*a*) *An engine working on Otto cycle, in which the salient points are 1, 2, 3 and 4, has upper and lower temperature limits T_3 and T_1. If the maximum work per kg of air is to be done, show that the intermediate temperature is given by*

$$T_2 = T_4 = \sqrt{T_1 T_3}\,.$$

(*b*) *If an engine works on Otto cycle between temperature limits 1450 K and 310 K, find the maximum power developed by the engine assuming the circulation of air per minute as 0.38 kg.*

Solution. (*a*) Refer Fig. 3.13 (Example 3.14).

Using the equation (*iii*) of example 3.14.

$$W = c_v\left[T_3 - T_1\,.\,(r)^{\gamma-1} - \frac{T_3}{(r)^{\gamma-1}} + T_1\right]$$

and differentiating W w.r.t. r and equating to zero

$$r = \left(\frac{T_3}{T_1}\right)^{1/2(\gamma-1)}$$

$$T_2 = T_1(r)^{\gamma-1} \text{ and } T_4 = T_3/(r)^{\gamma-1}$$

Substituting the value of r in the above equation

$$T_2 = T_1\left[\left(\frac{T_3}{T_1}\right)^{1/2(\gamma-1)}\right]^{\gamma-1} = T_1\left(\frac{T_3}{T_1}\right)^{1/2} = \sqrt{T_1 T_3}$$

Similarly,

$$T_4 = \frac{T_3}{\left[\left(\frac{T_3}{T_1}\right)^{1/2(\gamma-1)}\right]^{\gamma-1}} = \frac{T_3}{\left(\frac{T_3}{T_1}\right)^{1/2}} = \sqrt{T_3 T_1}$$

$\therefore \qquad T_2 = T_4 = \sqrt{T_1 T_3}$. **Proved.**

(*b*) **Power developed, P :**

$$\left.\begin{aligned} T_1 &= 310 \text{ K} \\ T_3 &= 1450 \text{ K} \\ m &= 0.38 \text{ kg} \end{aligned}\right\} \quad \text{.....(Given)}$$

Work done $\qquad W = c_v\ [(T_3 - T_2) - (T_4 - T_1)]$

$$T_2 = T_4 = \sqrt{T_1 T_3} = \sqrt{310 \times 1450} = 670.4 \text{ K}$$

$\therefore \qquad W = 0.71\ [(1450 - 670.4) - (670.4 - 310)]$

$\qquad = 0.71\ (779.6 - 360.4) = 297.6$ kJ/kg

Work done per second $\qquad = 297.6 \times (0.38/60) = 1.88$ kJ/s

Hence **power developed, P = 1.88 kW. (Ans.)**

Example 3.16. *For the same compression ratio, show that the efficiency of Otto cycle is greater than that of Diesel cycle.*

Solution. Refer Fig. 3.14.

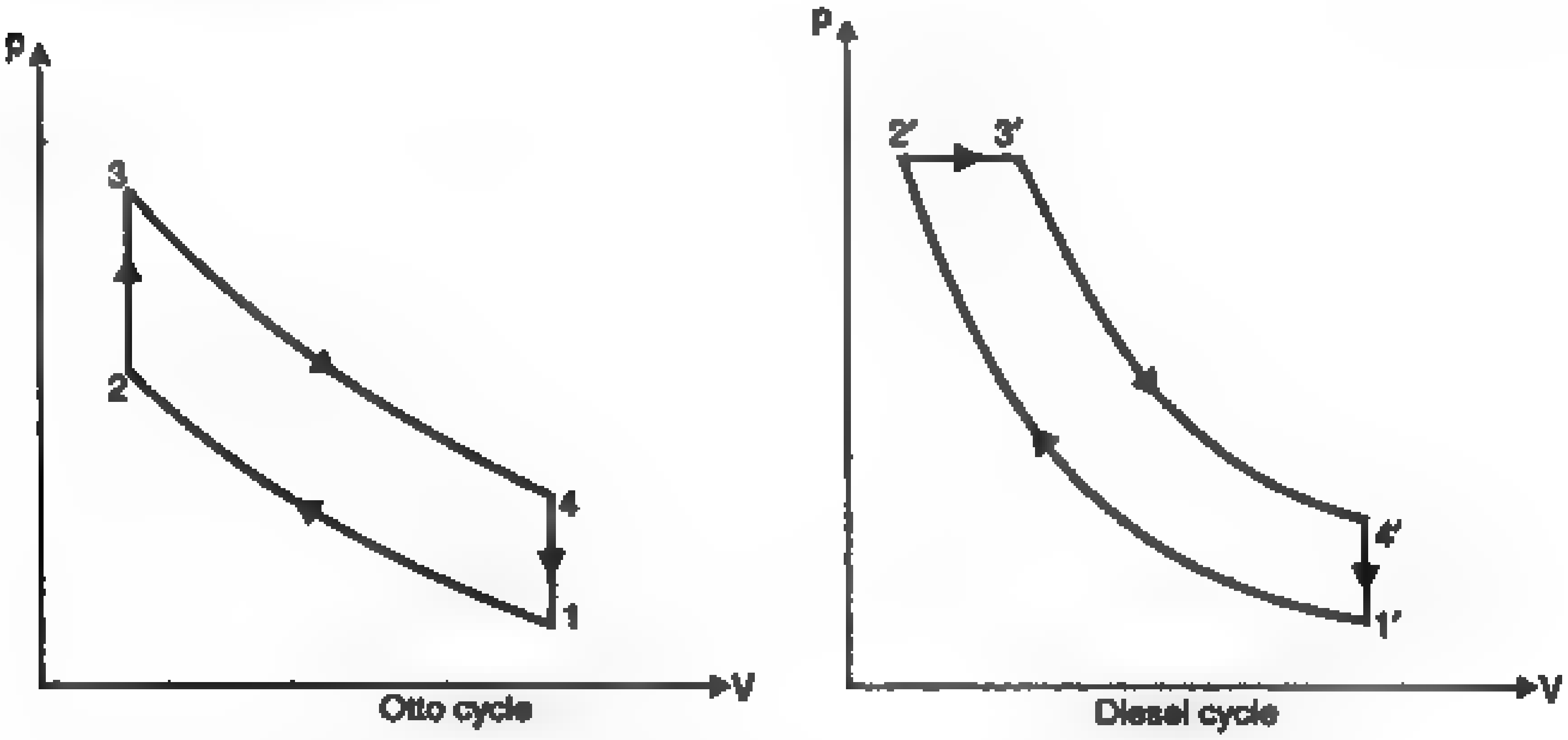


Fig. 3.14

We know that

$$\eta_{otto} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

and

$$\eta_{diesel} = 1 - \frac{1}{(r)^{\gamma - 1}} \times \frac{1}{\gamma} \left\{ \frac{\rho^{\gamma} - 1}{\rho - 1} \right\}$$

As the compression ratio is same,

$$\frac{V_1}{V_2} = \frac{V_1'}{V_2'} = r$$

If $\dfrac{V_4'}{V_3'} = r_1$, then cut off ratio, $\rho = \dfrac{V_3'}{V_2'} = \dfrac{r}{r_1}$

Putting the value of ρ in η_{diesel}, we get

$$\eta_{diesel} = 1 - \frac{1}{(r)^{\gamma-1}} \times \frac{1}{\gamma}\left[\frac{\left(\frac{r}{r_1}\right)^{\gamma} - 1}{\frac{r}{r_1} - 1}\right]$$

From above equation, we observe

$$\frac{r}{r_1} > 1$$

Let $r_1 = r - \delta$, where δ is a small quantity.

Then $$\frac{r}{r_1} = \frac{r}{r-\delta} = \frac{r}{r\left(1-\frac{\delta}{r}\right)} = \left(1-\frac{\delta}{r}\right)^{-1} = 1 + \frac{\delta}{r} + \frac{\delta^2}{r^2} + \frac{\delta^3}{r^3} + \ldots\ldots$$

and $$\left(\frac{r}{r_1}\right)^{\gamma} = \frac{r^{\gamma}}{r^{\gamma}\left(1-\frac{\delta}{r}\right)^{\gamma}} = \left(1-\frac{\delta}{r}\right)^{-\gamma} = 1 + \frac{\gamma\delta}{r} + \frac{\gamma(\gamma+1)}{2!}\cdot\frac{\delta^2}{r^2} + \ldots\ldots$$

$\therefore$ $$\eta_{diesel} = 1 - \frac{1}{(r)^{\gamma-1}} \times \frac{1}{\gamma}\left[\frac{\frac{\gamma\,.\,\delta}{r} + \frac{\gamma(\gamma+1)}{2!}\cdot\frac{\delta^2}{r^2} + \ldots\ldots}{\frac{\delta}{r} + \frac{\delta^2}{r^2} + \ldots\ldots}\right]$$

$$= 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{\frac{\delta}{r} + \frac{\gamma+1}{2}\cdot\frac{\delta^2}{r^2} + \ldots\ldots}{\frac{\delta}{r} + \frac{\delta^2}{r^2} + \ldots\ldots}\right]$$

The ratio inside the bracket is greater than 1 since the co-efficients of terms δ^2/r^2 is greater than 1 in the numerator. It means that something more is subtracted in case of diesel cycle than in Otto cycle.

Hence, *for same compression ratio* $\eta_{otto} > \eta_{diesel}$.

3.5. CONSTANT PRESSURE OR DIESEL CYCLE

This cycle was introduced by Dr. R. Diesel in 1897. It differs from Otto cycle, in, that heat is supplied at constant pressure instead of at constant volume. Fig. 3.15 (*a* and *b*) shows the *p-V* and *T-s* diagrams of this cycle respectively.

This cycle comprises of the following ***operations*** :

(*i*) 1-2......*Adiabatic compression.*

(*ii*) 2-3......*Addition of heat at constant pressure.*

(*iii*) 3-4......*Adiabatic expansion.*

(*iv*) 4-1......*Rejection of heat at constant volume.*

Point 1 represents that the cylinder is full of air. Let p_1, V_1 and T_1 be the corresponding pressure, volume and absolute temperature. The piston then compresses the air adiabatically (*i.e.* pV^{γ} = constant) till the values become p_2, V_2 and T_2 respectively (at the end of the stroke) at point 2. Heat is then added from a hot body at a constant pressure. During this addition of heat let

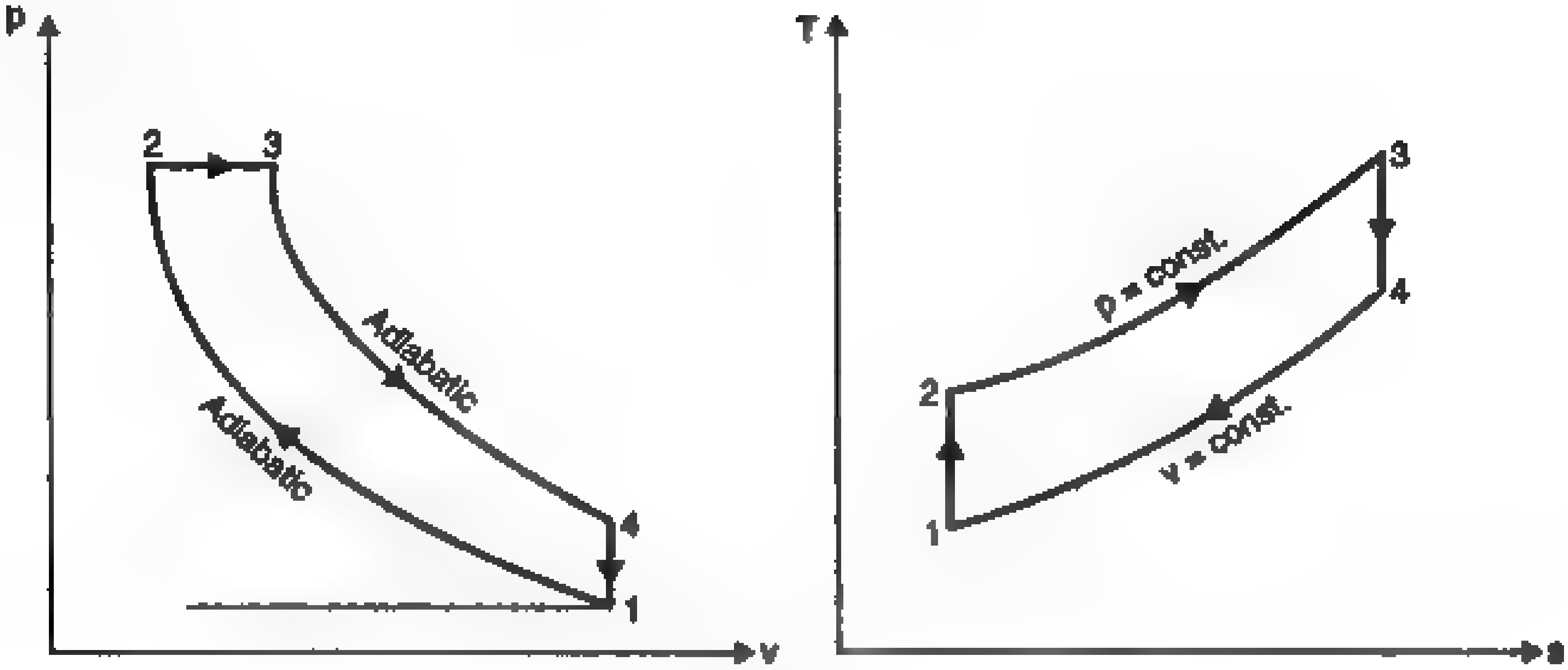


Fig. 3.15

volume increases from V_2 to V_3 and temperature T_2 to T_3, corresponding to point 3. This point (3) is called the *point of cut off*. The air then expands adiabatically to the conditions p_4, V_4 and T_4 respectively corresponding to point 4. Finally, the air rejects the heat to the cold body at constant volume till the point 1 where it returns to its original state.

Consider 1 kg of air.

Heat supplied at constant pressure $= c_p(T_3 - T_2)$

Heat rejected at constant volume $= c_v(T_4 - T_1)$

Work done = Heat supplied – Heat rejected

$$= c_p(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\therefore \qquad \eta_{diesel} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \qquad ...(i) \left[\because \frac{c_p}{c_v} = \gamma\right]$$

Let compression ratio, $r = \dfrac{v_1}{v_2}$ and cut off ratio, $\rho = \dfrac{v_3}{v_2}$ *i.e.* $\dfrac{\text{Volume at cut-off}}{\text{Clearance volume}}$

Now, during *adiabatic compression 1-2*,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \quad \text{or} \quad T_2 = T_1 \cdot (r)^{\gamma-1}$$

During *constant pressure process 2-3*,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \rho \quad \text{or} \quad T_3 = \rho \cdot T_2 = \rho \cdot T_1 \cdot (r)^{\gamma-1}$$

During *adiabatic expansion 3-4*,

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma-1} \qquad \left(\because \frac{v_4}{v_3} = \frac{v_1}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3} = \frac{r}{\rho}\right)$$

$$\therefore \qquad T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = \frac{\rho \,.\, T_1 (r)^{\gamma-1}}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = T_1 \,.\, \rho^{\gamma}$$

By inserting values of T_2, T_3 and T_4 in equation (i), we get

$$\eta_{\text{diesel}} = 1 - \frac{(T_1 \,.\, \rho^{\gamma} - T_1)}{\gamma(\rho \,.\, T_1 \,.\, (r)^{\gamma-1} - T_1 \,.\, (r)^{\gamma-1})} = 1 - \frac{(\rho^{\gamma} - 1)}{\gamma (r)^{\gamma-1} (\rho - 1)}$$

or

$$\eta_{\text{diesel}} = 1 - \frac{1}{\gamma (r)^{\gamma-1}} \left[\frac{\rho^{\gamma} - 1}{\rho - 1}\right] \qquad \text{...(3.7)}$$

It may be observed that equation (3.7) for efficiency of diesel cycle is different from that of the Otto cycle only in bracketed factor. This factor is always greater than unity, because $\rho > 1$. Hence *for a given compression ratio, the Otto cycle is more efficient*.

The *net work* for diesel cycle can be expressed in terms of pv as follows :

$$W = p_2(v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}$$

$$= p_2 (\rho v_2 - v_2) + \frac{p_3 \rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2 v_2 - p_1 r v_2}{\gamma - 1}$$

$$\left[\begin{array}{l} \because \dfrac{v_3}{v_2} = \rho \quad \therefore \; v_3 = \rho v_2 \text{ and } \dfrac{v_1}{v_2} = r \quad \therefore \; v_1 = r v_2 \\ \text{But } v_4 = v_1 \quad \therefore \quad v_4 = r v_2 \end{array}\right]$$

$$= p_2 v_2 (\rho - 1) + \frac{p_3 \rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2 v_2 - p_1 r v_2}{\gamma - 1}$$

$$= \frac{v_2 [p_2 (\rho - 1)(\gamma - 1) + p_3 \rho - p_4 r - (p_2 - p_1 r)]}{\gamma - 1}$$

$$= \frac{v_2 \left[p_2 (\rho - 1)(\gamma - 1) + p_3 \left(\rho - \frac{p_4 r}{p_3}\right) - p_2 \left(1 - \frac{p_1 r}{p_2}\right)\right]}{\gamma - 1}$$

$$= \frac{p_2 v_2 [(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma} \,.\, r^{1-\gamma} - (1 - r^{1-\gamma})]}{\gamma - 1}$$

$$\left[\because \frac{p_4}{p_3} = \left(\frac{v_3}{v_4}\right)^{\gamma} = \left(\frac{\rho}{r}\right)^{\gamma} = \rho^{\gamma} r^{-\gamma}\right]$$

$$= \frac{p_1 v_1 r^{\gamma-1} [(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma} r^{1-\gamma} - (1 - r^{1-\gamma})]}{\gamma - 1}$$

$$\left[\because \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} \text{ or } p_2 = p_1 \,.\, r^{\gamma} \text{ and } \frac{v_1}{v_2} = r \text{ or } v_2 = v_1 r^{-1}\right]$$

$$= \frac{p_1 v_1 r^{\gamma-1} [\gamma(\rho - 1) - r^{1-\gamma} (\rho^{\gamma} - 1)]}{(\gamma - 1)} \qquad \text{...(3.8)}$$

Mean effective pressure p_m is given by :

$$p_m = \frac{p_1 v_1 r^{\gamma-1}[\gamma(\rho-1) - r^{1-\gamma}(\rho^\gamma - 1)]}{(\gamma-1)v_1\left(\frac{r-1}{r}\right)}$$

or

$$\mathbf{p_m} = \frac{p_1 r^\gamma [\gamma(\rho-1) - r^{1-\gamma}(\rho^\gamma - 1)]}{(\gamma-1)(r-1)} \quad ...(3.9)$$

Example 3.17. *A diesel engine has a compression ratio of 15 and heat addition at constant pressure takes place at 6% of stroke. Find the air standard efficiency of the engine.*

Take γ for air as 1.4.

Solution. Refer Fig. 3.16.

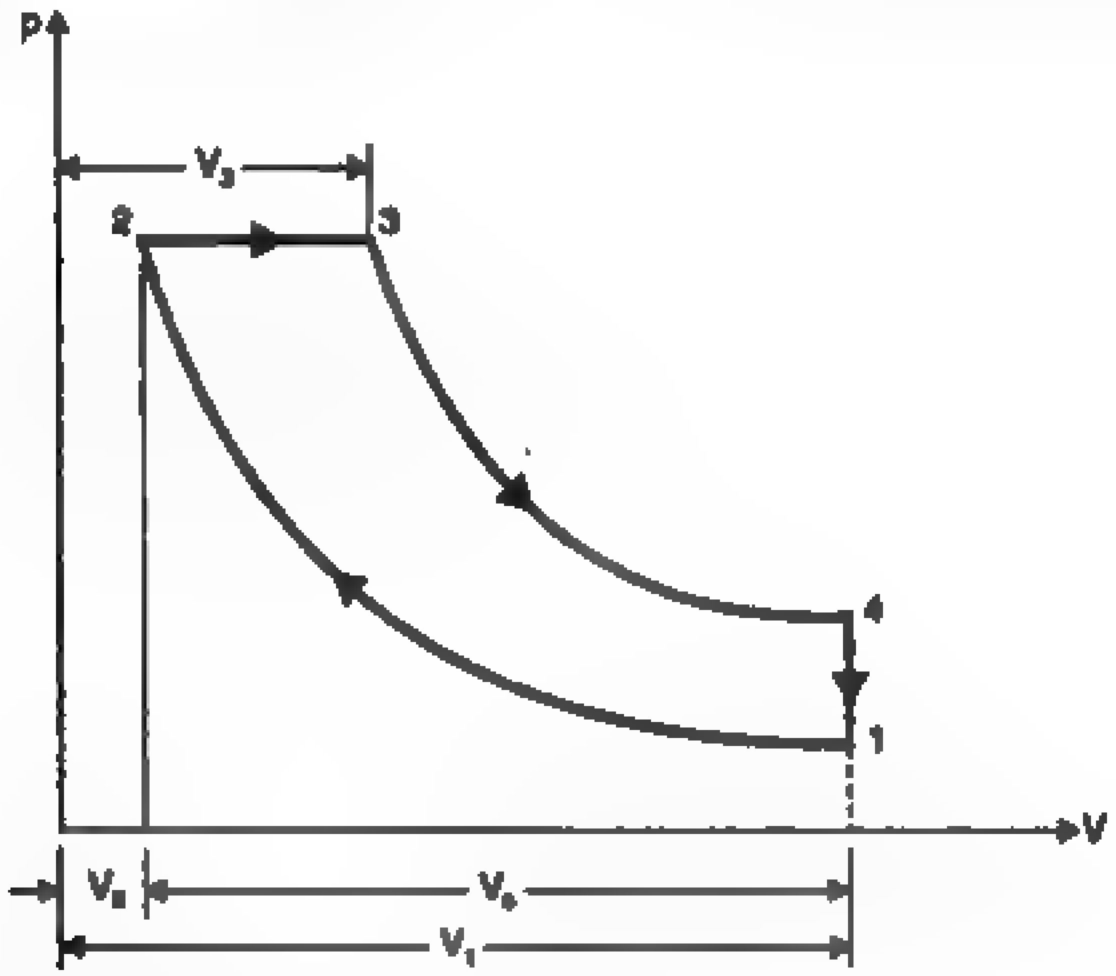


Fig. 3.16

Compression ratio, $r \left(= \frac{V_1}{V_2}\right) = 15$

γ for air = 1.4

Air standard efficiency of diesel cycle is given by

$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^\gamma - 1}{\rho - 1}\right] \quad ...(i)$$

where ρ = cut-off ratio = $\frac{V_3}{V_2}$

But $V_3 - V_2 = \frac{6}{100} V_s$ (V_s = stroke volume)

$$= 0.06\,(V_1 - V_2) = 0.06\,(15\,V_2 - V_2)$$
$$= 0.84\,V_2 \quad \text{or} \quad V_3 = 1.84\,V_2$$

$\therefore$ $$\rho = \frac{V_3}{V_2} = \frac{1.84\,V_2}{V_2} = 1.84$$

Putting the value in eqn. (*i*), we get

$$\eta_{diesel} = 1 - \frac{1}{1.4\,(15)^{1.4-1}}\left[\frac{(1.84)^{1.4}-1}{1.84-1}\right]$$
$$= 1 - 0.2417 \times 1.605 = \mathbf{0.612 \text{ or } 61.2\%.} \quad \textbf{(Ans.)}$$

Example 3.18. *The stroke and cylinder diameter of a compression ignition engine are 250 mm and 150 mm respectively. If the clearance volume is 0.0004 m³ and fuel injection takes place at constant pressure for 5 per cent of the stroke determine the efficiency of the engine. Assume the engine working on the diesel cycle.*

Solution. Refer Fig. 3.16.

Length of stroke, $L = 250 \text{ mm} = 0.25 \text{ m}$

Diameter of cylinder, $D = 150 \text{ mm} = 0.15 \text{ m}$

Clearance volume, $V_2 = 0.0004 \text{ m}^3$

Swept volume, $V_s = \pi/4\, D^2L = \pi/4 \times 0.15^2 \times 0.25 = 0.004418 \text{ m}^3$

Total cylinder volume $=$ Swept volume $+$ Clearance volume
$= 0.004418 + 0.0004 = 0.004818 \text{ m}^3$

Volume at point of cut-off, $$V_3 = V_2 + \frac{5}{100}\,V_s$$
$$= 0.0004 + \frac{5}{100} \times 0.004418 = 0.000621 \text{ m}^3$$

$\therefore$ Cut-off ratio, $$\rho = \frac{V_3}{V_2} = \frac{0.000621}{0.0004} = 1.55$$

Compression ratio, $$r = \frac{V_1}{V_2} = \frac{V_s + V_2}{V_2} = \frac{0.004418 + 0.0004}{0.0004} = 12.04$$

Hence, $$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^\gamma - 1}{\rho - 1}\right] = 1 - \frac{1}{1.4 \times (12.04)^{1.4-1}}\left[\frac{(1.55)^{1.4}-1}{1.55-1}\right]$$
$$= 1 - 0.264 \times 1.54 = \mathbf{0.593 \text{ or } 59.3\%.} \quad \textbf{(Ans.)}$$

Example 3.19. *Calculate the percentage loss in the ideal efficiency of a diesel engine with compression ratio 14 if the fuel cut-off is delayed from 5% to 8%.*

Solution. Let the clearance volume (V_2) be unity.

Then, compression ratio, $r = 14$

Now, when the fuel is cut-off at 5%, we have

$$\frac{\rho - 1}{r - 1} = \frac{5}{100} \quad \text{or} \quad \frac{\rho - 1}{14 - 1} = 0.05 \quad \text{or} \quad \rho - 1 = 13 \times 0.05 = 0.65$$

$\therefore$ $$\rho = 1.65$$

$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right] = 1 - \frac{1}{1.4 \times (14)^{1.4-1}}\left[\frac{(1.65)^{1.4}-1}{1.65-1}\right]$$

$$= 1 - 0.248 \times 1.563 = 0.612 \quad \text{or} \quad 61.2\%$$

When the fuel is cut-off at 8%, we have

$$\frac{\rho-1}{r-1} = \frac{8}{100} \quad \text{or} \quad \frac{\rho-1}{14-1} = \frac{8}{100} = 0.08$$

$$\therefore \qquad \rho = 1 + 1.04 = 2.04$$

$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right] = 1 - \frac{1}{1.4 \times (14)^{1.4-1}}\left[\frac{(2.04)^{1.4}-1}{2.04-1}\right]$$

$$= 1 - 0.248 \times 1.647 = 0.591 \quad \text{or} \quad 59.1\%. \quad \text{(Ans.)}$$

Hence percentage loss in efficiency due to delay in fuel cut-off

$$= 61.2 - 59.1 = 2.1\%. \quad \text{(Ans.)}$$

Example 3.20. *The mean effective pressure of a Diesel cycle is 7.5 bar and compression ratio is 12.5. Find the percentage cut-off of the cycle if its initial pressure is 1 bar.*

Solution. Mean effective pressure, $p_m = 7.5$ bar

Compression ratio, $r = 12.5$

Initial pressure, $p_1 = 1$ bar

Refer Fig. 3.15.

The mean effective pressure is given by

$$p_m = \frac{p_1 r^{\gamma}[\gamma(\rho-1) - r^{1-\gamma}(\rho^{\gamma}-1)]}{(\gamma-1)(r-1)} \qquad \text{...[Eqn. (3.9)]}$$

$$7.5 = \frac{1 \times (12.5)^{1.4}[1.4(\rho-1) - (12.5)^{1-1.4}(\rho^{1.4}-1)]}{(1.4-1)(12.5-1)}$$

$$7.5 = \frac{34.33[1.4\rho - 1.4 - 0.364\rho^{1.4} + 0.364]}{4.6}$$

$$7.5 = 7.46\,(1.4\rho - 1.036 - 0.364\,\rho^{1.4})$$

$$1.005 = 1.4\,\rho - 1.036 - 0.364\,\rho^{1.4}$$

or

$$2.04 = 1.4\,\rho - 0.364\,\rho^{1.4} \quad \text{or} \quad 0.346\,\rho^{1.4} - 1.4\,\rho + 2.04 = 0$$

Solving by trial and error method, we get

$$\rho = 2.24$$

$$\therefore \qquad \%\ \textit{cut-off} = \frac{\rho-1}{r-1} \times 100 = \frac{2.24-1}{12.5-1} \times 100 = 10.78\%. \quad \text{(Ans.)}$$

Example 3.21. *An engine with 200 mm cylinder diameter and 300 mm stroke works on theoretical Diesel cycle. The initial pressure and temperature of air used are 1 bar and 27°C. The cut-off is 8% of the stroke. Determine :*

(i) Pressures and temperatures at all salient points.

(ii) Theoretical air standard efficiency.

(iii) Mean effective pressure.

(iv) Power of the engine if the working cycles per minute are 380.

Assume that compression ratio is 15 and working fluid is air.

Consider all conditions to be ideal.

Solution. Refer Fig. 3.17.

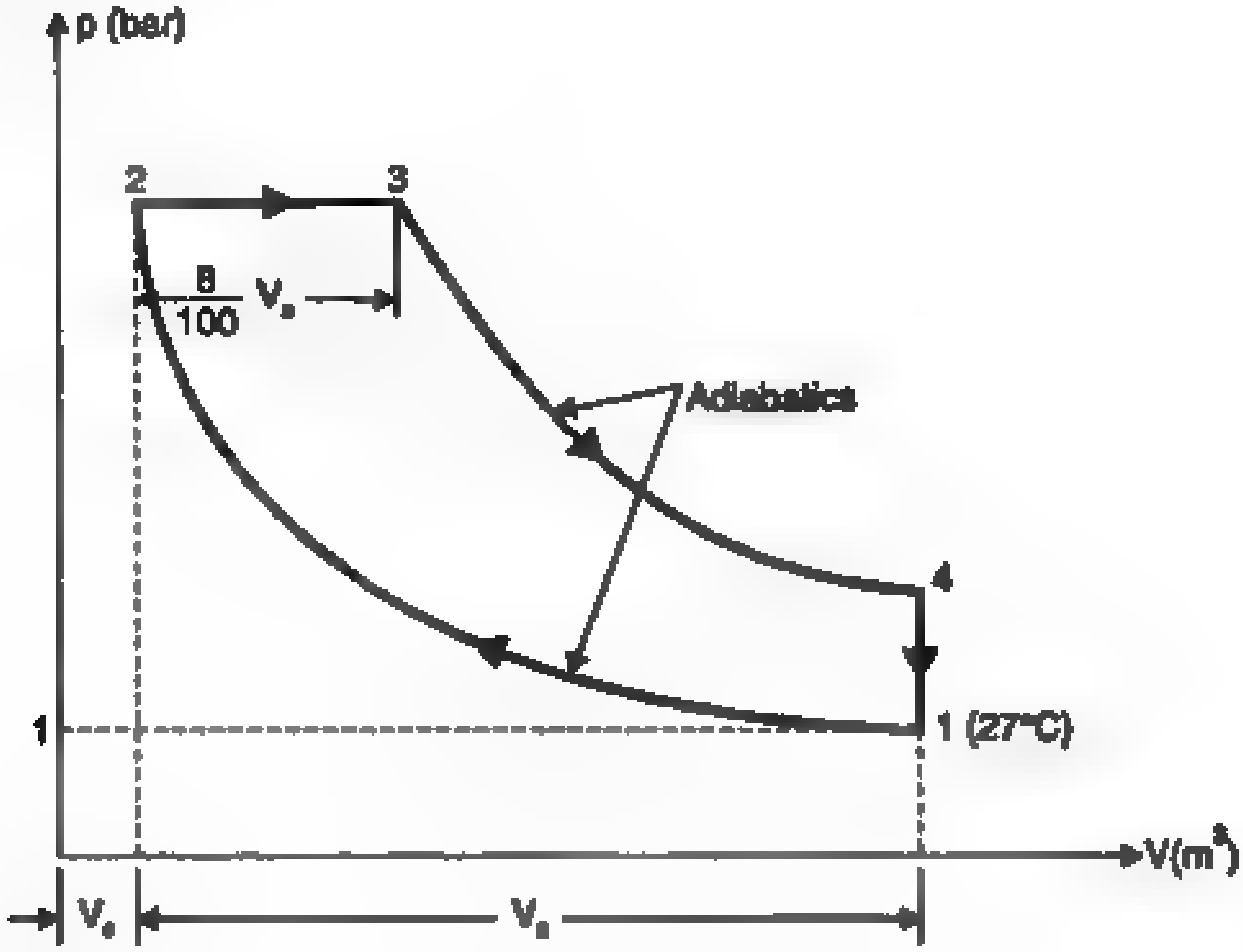


Fig. 3.17

Cylinder diameter, $D = 200$ mm or 0.2 m

Stroke length, $L = 300$ mm or 0.3 m

Initial pressure, $p_1 = 1.0$ bar

Initial temperature, $T_1 = 27 + 273 = 300$ K

Cut-off $= \dfrac{8}{100} V_s = 0.08\, V_s$

(i) Pressures and temperatures at salient points :

Now, stroke volume, $V_s = \pi/4\, D^2L = \pi/4 \times 0.2^2 \times 0.3 = 0.00942$ m³

$$V_1 = V_s + V_c = V_s + \frac{V_s}{r-1} \qquad \left[\because V_c = \frac{V_s}{r-1}\right]$$

$$= V_s\left(1 + \frac{1}{r-1}\right) = \frac{r}{r-1} \times V_s$$

i.e., $$V_1 = \frac{15}{15-1} \times V_s = \frac{15}{14} \times 0.00942 = \mathbf{0.0101\ m^3.} \quad \textbf{(Ans.)}$$

Mass of the air in the cylinder can be calculated by using the gas equation,

$$p_1V_1 = mRT_1$$

$$m = \frac{p_1V_1}{RT_1} = \frac{1 \times 10^5 \times 0.0101}{287 \times 300} = 0.0117 \text{ kg/cycle}$$

For the *adiabatic (or isentropic) process 1-2,*

$$p_1V_1^{\gamma} = p_2V_2^{\gamma} \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = (r)^{\gamma}$$

$\therefore$ $\mathbf{p_2} = p_1 \cdot (r)^{\gamma} = 1 \times (15)^{1.4} = \mathbf{44.31\ bar.}$ **(Ans.)**

Also, $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (15)^{1.4-1} = 2.954$$

$\therefore$ $\mathbf{T_2} = T_1 \times 2.954 = 300 \times 2.954 = \mathbf{886.2\ K.}$ **(Ans.)**

$$\mathbf{V_2} = V_c = \frac{V_s}{r-1} = \frac{0.00942}{15-1} = \mathbf{0.0006728\ m^3.}\ \textbf{(Ans.)}$$

$\mathbf{p_3} = \mathbf{p_2} = \mathbf{44.31\ bar.}$ **(Ans.)**

% cut-off ratio $= \dfrac{\rho - 1}{r-1}$

$$\frac{8}{100} = \frac{\rho-1}{15-1}$$

i.e., $\rho = 0.08 \times 14 + 1 = 2.12$

$\therefore$ $\mathbf{V_3} = \rho\, V_2 = 2.12 \times 0.0006728 = \mathbf{0.001426\ m^3.}$ **(Ans.)**

$$\left[\begin{array}{l} V_3 \text{ can also be calculated as follows:} \\ V_3 = 0.08V_s + V_c = 0.08 \times 0.00942 + 0.0006728 = 0.001426\ \text{m}^3 \end{array}\right]$$

For the *constant pressure process 2-3,*

$$\frac{V_2}{T_2} = \frac{V_3}{T_3}$$

$$\therefore \quad \mathbf{T_3} = T_2 \times \frac{V_3}{V_2} = 886.2 \times \frac{0.001426}{0.0006728} = \mathbf{1878.3\ K.}\ \textbf{(Ans.)}$$

For the *isentropic process 3-4*

$$p_3V_3^{\gamma} = p_4V_4^{\gamma}$$

$$\mathbf{p_4} = p_3 \times \left(\frac{V_3}{V_4}\right)^{\gamma} = p_3 \times \frac{1}{(7.07)^{1.4}} \qquad \left[\because\ \frac{V_4}{V_3} = \frac{V_4}{V_2} \times \frac{V_2}{V_3} = \frac{V_1}{V_2} \times \frac{V_2}{V_3} = \frac{r}{\rho},\ \because\ V_4 = V_1 = \frac{15}{2.12} = 7.07\right]$$

$$= \frac{44.31}{(7.07)^{1.4}} = \mathbf{2.866\ bar.}\ \textbf{(Ans.)}$$

Also, $$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{1}{7.07}\right)^{1.4-1} = 0.457$$

$\therefore$ $\mathbf{T_4} = T_3 \times 0.457 = 1878.3 \times 0.457 = \mathbf{858.38\ K.}$ **(Ans.)**

$\mathbf{V_4} = V_1 = \mathbf{0.0101\ m^3.}$ **(Ans.)**

(*ii*) **Theoretical air standard efficiency :**

$$\eta_{diesel} = 1 - \frac{1}{\gamma(r)^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\rho-1}\right] = 1 - \frac{1}{1.4\,(15)^{1.4-1}}\left[\frac{(2.12)^{1.4}-1}{2.12-1}\right]$$

$$= 1 - 0.2418 \times 1.663 = \mathbf{0.598\ or\ 59.8\%.}\ \textbf{(Ans.)}$$

(*iii*) **Mean effective pressure, p_m :**

Mean effective pressure of Diesel cycle is given by

$$p_m = \frac{p_1(r)^\gamma[\gamma(\rho-1)-r^{1-\gamma}(\rho^\gamma-1)]}{(\gamma-1)(r-1)}$$

$$= \frac{1\times(15)^{1.4}[1.4(2.12-1)-(15)^{1-1.4}(2.12^{1.4}-1)]}{(1.4-1)(15-1)}$$

$$= \frac{44.31[1.568-0.338\times1.863]}{0.4\times14} = 7.424 \text{ bar. (Ans.)}$$

(*iv*) **Power of the engine, P :**

Work done per cycle $= p_mV_s = \dfrac{7.424\times10^5\times0.00942}{10^3} = 6.99$ kJ/cycle

Work done per second = Work done per cycle × No. of cycles per second
$= 6.99 \times 380/60 = 44.27$ kJ/s $= 44.27$ kW

Hence *power of the engine* = **44.27 kW. (Ans.)**

Example 3.22. *The volume ratios of compression and expansion for a diesel engine as measured from an indicator diagram are 15.3 and 7.5 respectively. The pressure and temperature at the beginning of the compression are 1 bar and 27°C.*

Assuming an ideal engine, determine the mean effective pressure, the ratio of maximum pressure to mean effective pressure and cycle efficiency.

Also find the fuel consumption per kWh if the indicated thermal efficiency is 0.5 of ideal efficiency, mechanical efficiency is 0.8 and the calorific value of oil 42000 kJ/kg.

Assume for air : $c_p = 1.005$ kJ/kg K ; $c_v = 0.718$ kJ/kg K, $\gamma = 1.4$. **(U.P.S.C. 1996)**

Solution. Refer Fig. 3.18. *Given :* $\dfrac{V_1}{V_2} = 15.3$; $\dfrac{V_4}{V_3} = 7.5$

$p_1 = 1$ bar ; $T_1 = 27 + 273 = 300$ K ; $\eta_{th(I)} = 0.5 \times \eta_{air\text{-}standard}$; $\eta_{mech.} = 0.8$; $C = 42000$ kJ/kg.

The cycle is shown in the Fig. 3.18, the subscripts denote the respective points in the cycle.

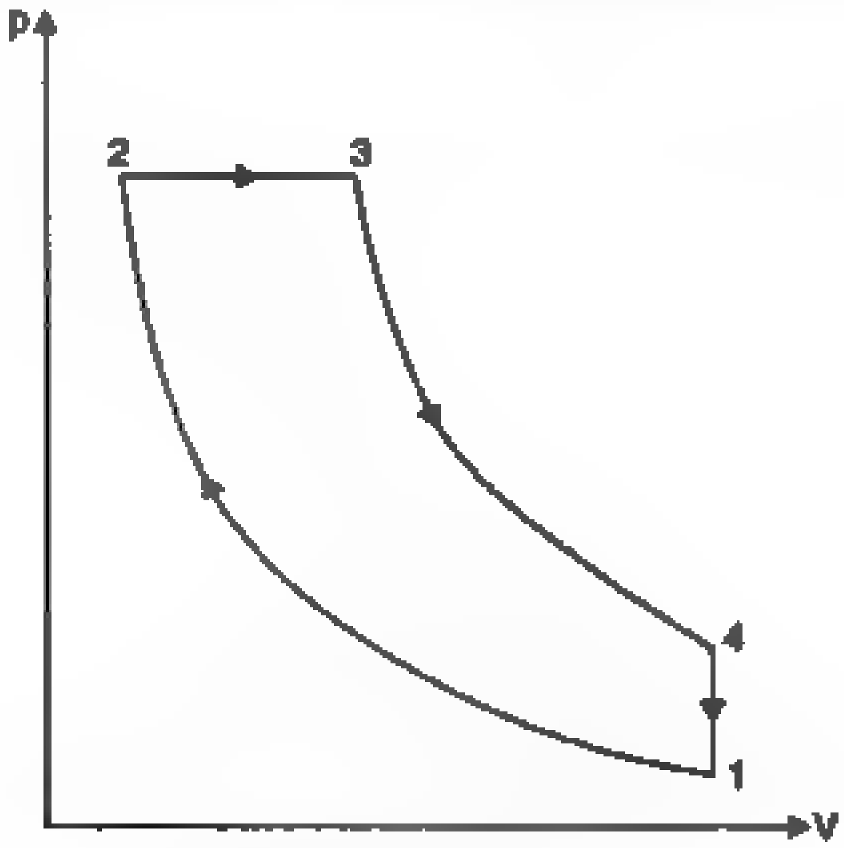


Fig. 3.18. Diesel cycle.

Mean effective pressure, p_m :

$$p_m = \frac{\text{Work done by the cycle}}{\text{Swept volume}}$$

Work done = Heat added – Heat rejected

Heat added $= mc_p\,(T_3 - T_2)$, and

Heat rejected $= mc_v\,(T_4 - T_1)$

Now assume air as a perfect gas and mass of oil in the air-fuel mixture is negligible and is not taken into account.

Process 1-2 is an adiabatic compression process, thus

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad \text{or} \quad T_2 = T_1 \times \left(\frac{V_1}{V_2}\right)^{1.4-1} \quad (\text{since } \gamma = 1.4)$$

or $$T_2 = 300 \times (15.3)^{0.4} = 893.3 \text{ K}$$

Also, $$p_1V_1^{\gamma} = p_2V_2^{\gamma} \Rightarrow p_2 = p_1 \times \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times (15.3)^{1.4} = 45.56 \text{ bar}$$

Process 2-3 is a constant pressure process, hence

$$\frac{V_2}{T_2} = \frac{V_3}{T_3} \Rightarrow T_3 = \frac{V_3T_2}{V_2} = 2.04 \times 893.3 = 1822.3 \text{ K}$$

Assume that the volume at point 2 (V_2) is 1 m³. Thus the mass of air involved in the process,

$$m = \frac{p_2V_2}{RT_2} = \frac{45.56 \times 10^5 \times 1}{287 \times 893.3} = 17.77 \text{ kg} \qquad \left[\because \ \frac{V_4}{V_3} = \frac{V_1}{V_3} = \frac{V_1}{V_2} \times \frac{V_2}{V_3} \quad \text{or} \quad \frac{V_3}{V_2} = \frac{V_1}{V_2} \times \frac{V_3}{V_4} = \frac{15.3}{7.5} = 2.04\right]$$

Process 3-4 is an adiabatic expansion process, thus

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{1}{7.5}\right)^{1.4-1} = 0.4466$$

or $$T_4 = 1822.3 \times 0.4466 = 813.8 \text{ K}$$

$\therefore$ Work done $= mc_p\,(T_3 - T_2) - mc_v\,(T_4 - T_1)$

$= 17.77\,[1.005\,(1822.3 - 893.3) - 0.718\,(813.8 - 300)] = 10035$ kJ

$$\therefore \quad p_m = \frac{\text{Work done}}{\text{Swept volume}} = \frac{10035}{(V_1 - V_2)} = \frac{10035}{(15.3V_2 - V_2)} = \frac{10035}{14.3}$$

$= 701.7$ kN/m² $= 7.017$ bar. **(Ans.)**

($\because$ $V_2 = 1$ m³ assumed)

Ratio of maximum pressure to mean effective pressure

$$= \frac{p_2}{p_m} = \frac{45.56}{7.017} = 6.49. \quad \textbf{(Ans.)}$$

Cycle efficiency, η_{cycle} :

$$\eta_{cycle} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{10035}{mc_p\,(T_3 - T_2)} = \frac{10035}{17.77 \times 1.005\,(1822.3 - 897.3)} = \mathbf{0.6048 \quad or \quad 60.48\%.} \quad \textbf{(Ans.)}$$

Fuel consumption per kWh ; m_f

$\eta_{th(T)} = 0.5\ \eta_{cycle} = 0.5 \times 0.6048 = 0.3024$ or 30.24%

$$\eta_{th(B)} = 0.3024 \times 0.8 = 0.242$$

Also,
$$\eta_{th(B)} = \frac{\text{B.P.}}{m_f \times C} = \frac{1}{\frac{m_f}{3600} \times 42000} = \frac{3600}{m_f \times 42000}$$

or
$$0.242 = \frac{3600}{m_f \times 42000}$$

or
$$m_f = \frac{3600}{0.242 \times 42000} = \mathbf{0.354\ kg/kWh.} \quad \textbf{(Ans.)}$$

3.6. DUAL COMBUSTION CYCLE

This cycle (also called the *limited pressure cycle or mixed cycle*) is a combination of Otto and Diesel cycles, in a way, that *heat is added partly at constant volume and partly at constant pressure ; the advantage of which is that more time is available to fuel (which is injected into the engine cylinder before the end of compression stroke) for combustion. Because of lagging characteristics of fuel this cycle is invariably used for diesel and hot spot ignition engines.*

The dual combustion cycle (Fig. 3.19) consists of the following ***operations*** :

(*i*) 1-2—*Adiabatic compression*

(*ii*) 2-3—*Addition of heat at constant volume*

(*iii*) 3-4—*Addition of heat at constant pressure*

(*iv*) 4-5—*Adiabatic expansion*

(*v*) 5-1—*Rejection of heat at constant volume.*

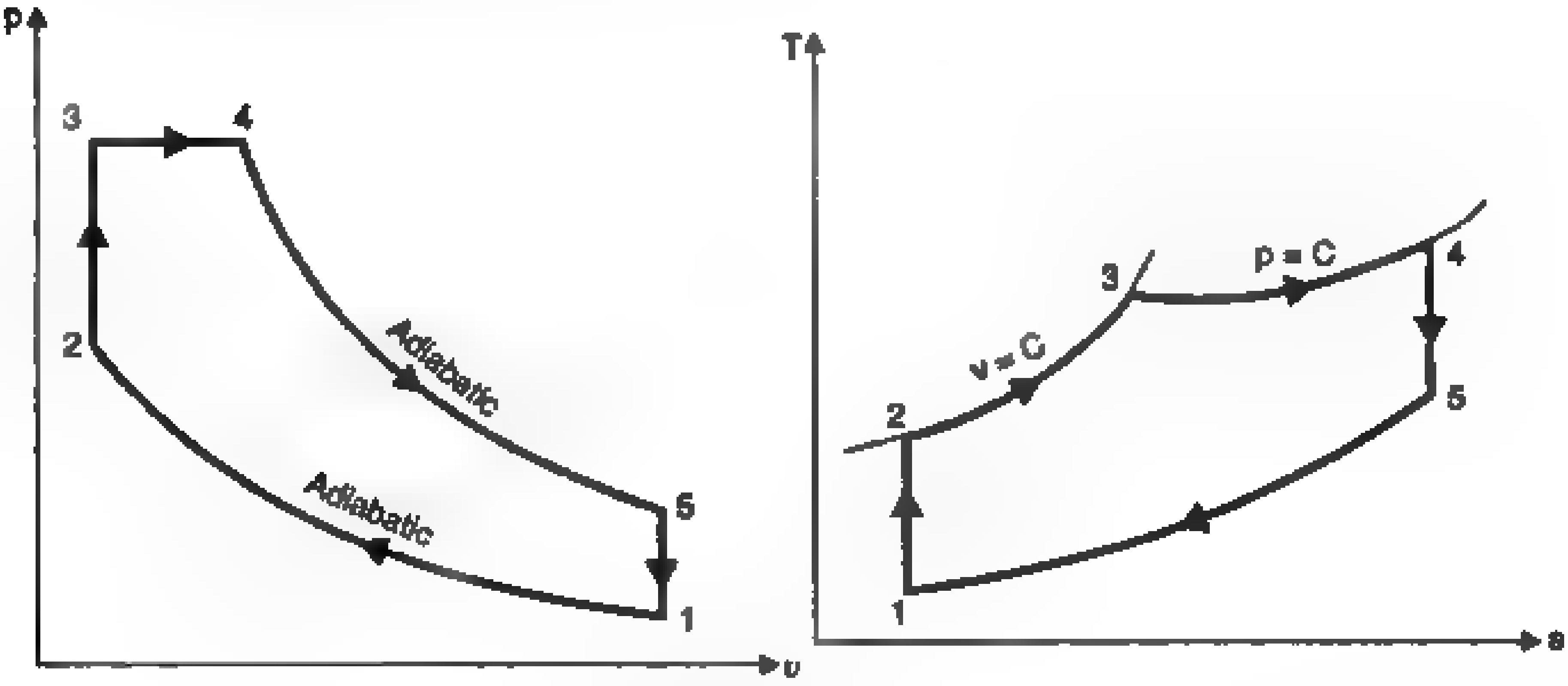


Fig. 3.19

Consider 1 kg of air.

Total heat supplied $=$ Heat supplied during the operation 2-3

$\qquad +$ Heat supplied during the operation 3-4

$= c_v(T_3 - T_2) + c_p(T_4 - T_3)$

Heat rejected during operation 5-1 $= c_v(T_5 - T_1)$

Work done $=$ Heat supplied $-$ Heat rejected

$= c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)$

$$\eta_{dual} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v\,(T_3 - T_2) + c_p\,(T_4 - T_3) - c_v\,(T_5 - T_1)}{c_v\,(T_3 - T_2) + c_p\,(T_4 - T_3)}$$

$$= 1 - \frac{c_v\,(T_5 - T_1)}{c_v\,(T_3 - T_2) + c_p\,(T_4 - T_3)}$$

$$= 1 - \frac{c_v\,(T_5 - T_1)}{(T_3 - T_2) + \gamma\,(T_4 - T_3)} \qquad \text{...}(i) \qquad \left(\because \ \gamma = \frac{c_p}{c_v}\right)$$

Compression ratio, $r = \dfrac{v_1}{v_2}$

During *adiabatic compression process 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} \qquad \text{...}(ii)$$

During *constant volume heating process,*

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

or
$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \beta, \text{ where } \beta \text{ is known as } \textbf{\textit{pressure or explosion ratio.}}$$

or
$$T_2 = \frac{T_3}{\beta} \qquad \text{...}(iii)$$

During *adiabatic expansion process 4-5,*

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma - 1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma - 1} \qquad \text{...}(iv)$$

$$\left(\because \ \frac{v_5}{v_4} = \frac{v_1}{v_4} = \frac{v_1}{v_2} \times \frac{v_2}{v_4} = \frac{v_1}{v_2} \times \frac{v_3}{v_4} = \frac{r}{\rho}, \rho \text{ being the cut-off ratio}\right)$$

During *constant pressure heating process 3-4,*

$$\frac{v_3}{T_3} = \frac{v_4}{T_4}$$

$$T_4 = T_3\,\frac{v_4}{v_3} = \rho\,T_3 \qquad \text{...}(v)$$

Putting the value of T_4 in the equation (iv), we get

$$\frac{\rho T_3}{T_5} = \left(\frac{r}{\rho}\right)^{\gamma - 1} \quad \text{or} \quad T_5 = \rho\,.\,T_3\,.\left(\frac{\rho}{\gamma}\right)^{\gamma - 1}$$

Putting the value of T_2 in equation (*ii*), we get

$$\frac{\frac{T_3}{\beta}}{T_1} = (r)^{\gamma-1}$$

$$T_1 = \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma-1}}$$

Now inserting the values of T_1, T_2, T_4 and T_5 in equation (*i*), we get

$$\eta_{dual} = 1 - \frac{\left[\rho \cdot T_3\left(\frac{\rho}{r}\right)^{\gamma-1} - \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma-1}}\right]}{\left[\left(T_3 - \frac{T_3}{\beta}\right) + \gamma(\rho T_3 - T_3)\right]} = 1 - \frac{\frac{1}{(r)^{\gamma-1}}\left(\rho^\gamma - \frac{1}{\beta}\right)}{\left(1 - \frac{1}{\beta}\right) + \gamma(\rho - 1)}$$

i.e.,

$$\eta_{dual} = 1 - \frac{1}{(r)^{\gamma-1}} \cdot \frac{(\beta \cdot \rho^\gamma - 1)}{[(\beta - 1) + \beta\gamma(\rho - 1)]} \qquad ...(3.10)$$

Work done is given by,

$$W = p_3(v_4 - v_3) + \frac{p_4v_4 - p_5v_5}{\gamma - 1} - \frac{p_2v_2 - p_1v_1}{\gamma - 1}$$

$$= p_3v_3(\rho - 1) + \frac{(p_4\rho v_3 - p_5rv_3) - (p_2v_3 - p_1rv_3)}{\gamma - 1}$$

$$= \frac{p_3v_3(\rho - 1)(\gamma - 1) + p_4v_3\left(\rho - \frac{p_5}{p_4}r\right) - p_2v_3\left(1 - \frac{p_1}{p_2}r\right)}{\gamma - 1}$$

Also $\qquad \frac{p_5}{p_4} = \left(\frac{v_4}{v_5}\right)^\gamma = \left(\frac{\rho}{r}\right)^\gamma$ and $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma = r^\gamma$

also, $\qquad p_3 = p_4,\ v_2 = v_3,\ v_5 = v_1$

$$\therefore \qquad W = \frac{v_3[p_3(\rho - 1)(\gamma - 1) + p_3(\rho - \rho^\gamma r^{1-\gamma}) - p_2(1 - r^{1-\gamma})]}{(\gamma - 1)}$$

$$= \frac{p_2v_2[\beta(\rho - 1)(\gamma - 1) + \beta(\rho - \rho^\gamma r^{1-\gamma}) - (1 - r^{1-\gamma})]}{(\gamma - 1)}$$

$$= \frac{p_1(r)^\gamma v_1 / r[\beta\gamma(\rho - 1) + (\beta - 1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{\gamma - 1}$$

$$= \frac{p_1v_1r^{\gamma-1}[\beta\gamma(\rho - 1) + (\beta - 1) - r^{\gamma-1}(\beta\rho^\gamma - 1)]}{\gamma - 1} \qquad ...(3.11)$$

Mean effective pressure (p_m) is given by,

$$p_m = \frac{W}{v_1 - v_2} = \frac{W}{v_1\left(\frac{r-1}{r}\right)} = \frac{p_1v_1[r^{1-\gamma}\beta\gamma(\rho - 1) + (\beta - 1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{(\gamma - 1)v_1\left(\frac{r-1}{r}\right)}$$

$$\mathbf{p_m} = \frac{p_1(r)^\gamma[\beta(\rho - 1) + (\beta - 1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{(\gamma - 1)(r - 1)} \qquad ...(3.12)$$

Example 3.23. *The swept volume of a diesel engine working on dual cycle is 0.0053 m^3 and clearance volume is 0.00035 m^3. The maximum pressure is 65 bar. Fuel injection ends at 5 per cent of the stroke. The temperature and pressure at the start of the compression are 80°C and 0.9 bar. Determine the air standard efficiency of the cycle. Take γ for air = 1.4.*

Solution. Refer Fig. 3.20.

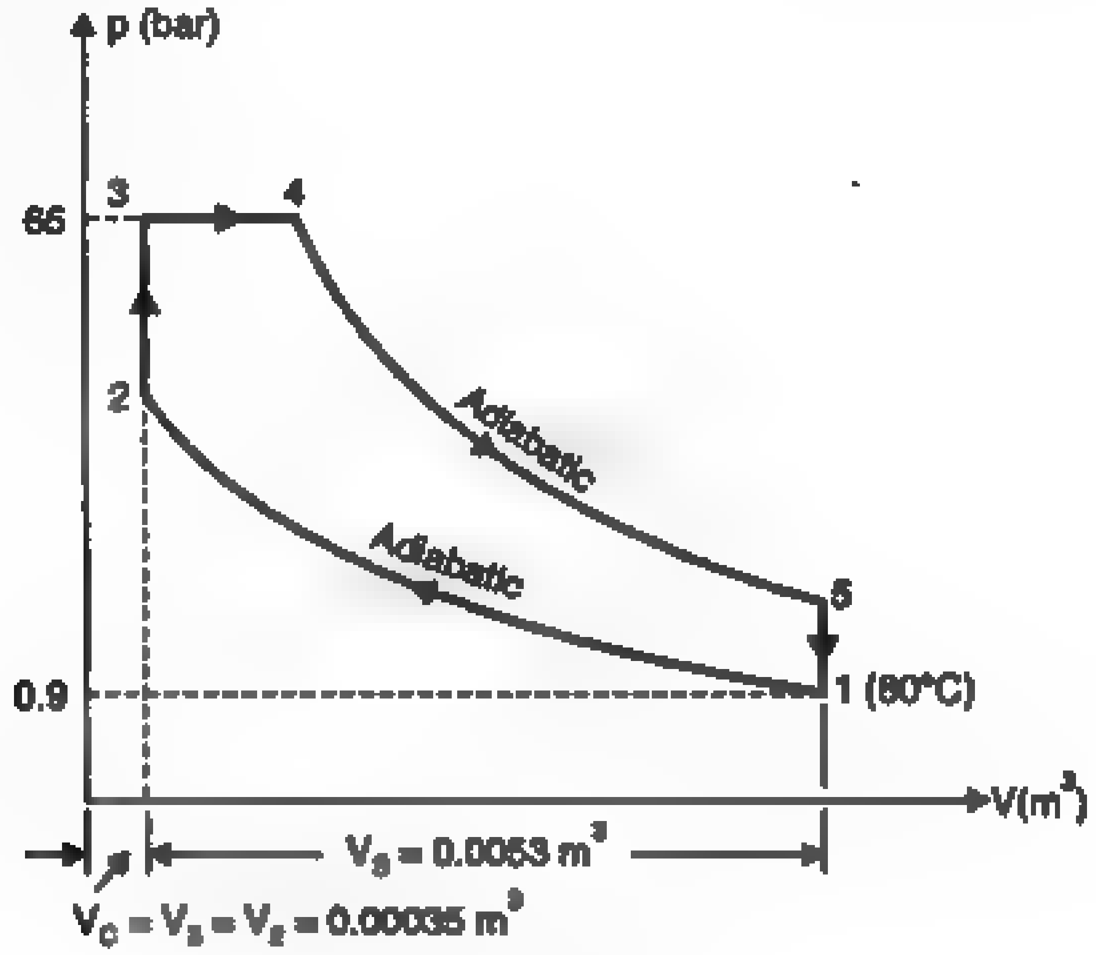


Fig. 3.20

Swept volume, $V_s = 0.0053\ m^3$

Clearance volume, $V_c = V_3 = V_2 = 0.00035\ m^3$

Maximum pressure, $p_3 = p_4 = 65$ bar

Initial temperature, $T_1 = 80 + 273 = 353$ K

Initial pressure, $p_1 = 0.9$ bar

$\eta_{dual} = ?$

The efficiency of a dual combustion cycle is given by

$$\eta_{dual} = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{\beta \cdot \rho^{\gamma} - 1}{(\beta - 1) + \beta\gamma(\rho - 1)}\right] \quad ...(i)$$

Compression ratio, $r = \frac{V_1}{V_2} = \frac{V_s + V_c}{V_c} = \frac{0.0053 + 0.00035}{0.00035} = 16.14$

[$\because$ $V_2 = V_c$ = Clearance volume]

Cut-off ratio, $\rho = \frac{V_4}{V_3} = \frac{\frac{5}{100}V_s + V_3}{V_3} = \frac{0.05V_s + V_c}{V_c}$ ($\because$ $V_2 = V_3 = V_c$)

$$= \frac{0.05 \times 0.0053 + 0.00035}{0.00035} = 1.757 \text{ say } 1.76$$

Also during the *compression operation 1-2*

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

or
$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = (16.14)^{1.4} = 49.14$$

or
$$p_2 = p_1 \times 49.14 = 0.9 \times 49.14 = 44.22 \text{ bar}$$

Pressure or explosion ratio, $\beta = \frac{p_3}{p_2} = \frac{65}{44.22} = 1.47$

Putting the value of r, ρ and β in equation (*i*), we get

$$\eta_{dual} = 1 - \frac{1}{(16.14)^{1.4-1}}\left[\frac{1.47 \times (1.76)^{1.4} - 1}{(1.47 - 1) + 1.47 \times 1.4\,(1.76 - 1)}\right]$$

$$= 7 - 0.328\left[\frac{3.243 - 1}{0.47 + 1.564}\right] = \mathbf{0.6383 \text{ or } 63.83\%.} \quad \textbf{(Ans.)}$$

Example 3.24. *An oil engine working on the dual combustion cycle has a compression ratio 14 and the explosion ratio obtained from an indicator card is 1.4. If the cut-off occurs at 6 per cent of stroke, find the ideal efficiency. Take γ for air = 1.4.*

Solution. Refer Fig. 3.19.

Compression ratio, $r = 14$

Explosion ratio, $\beta = 1.4$

If ρ is the cut-off ratio, then $\frac{\rho - 1}{r - 1} = \frac{6}{100}$ or $\frac{\rho - 1}{14 - 1} = 0.06$

$\therefore$ $\rho = 1.78$

Ideal efficiency is given by

$$\eta_{ideal\ or\ dual} = 1 - \frac{1}{(r)^{\gamma-1}}\left[\frac{(\beta\rho^{\gamma} - 1)}{(\beta - 1) + \beta\gamma(\rho - 1)}\right]$$

$$= 1 - \frac{1}{(14)^{1.4-1}}\left[\frac{1.4 \times (1.78)^{1.4} - 1}{(1.4 - 1) + 1.4 \times 1.4(1.78 - 1)}\right]$$

$$= 1 - 0.348\left[\frac{3.138 - 1}{0.4 + 1.528}\right] = \mathbf{0.614 \text{ or } 61.4\%.} \quad \textbf{(Ans.)}$$

Example 3.25. *The compression ratio for a single-cylinder engine operating on dual cycle is 9. The maximum pressure in the cylinder is limited to 60 bar. The pressure and temperature of the air at the beginning of the cycle are 1 bar and 30°C. Heat is added during constant pressure process upto 4 per cent of the stroke. Assuming the cylinder diameter and stroke length as 250 mm and 300 mm respectively, determine :*

(*i*) *The air standard efficiency of the cycle.*

(*ii*) *The power developed if the number of working cycles are 3 per second.*

Take for air $c_v = 0.71$ *kJ/kg K and :* $c_p = 1.0$ *kJ/kgK.*

Solution. Refer Fig. 3.21.

Cylinder diameter, $D = 250 \text{ mm} = 0.25 \text{ m}$

Compression ratio, $r = 9$

Stroke length, $L = 300 \text{ mm} = 0.3 \text{ m}$

Initial pressure, $p_1 = 1$ bar
Initial temperature, $T_1 = 30 + 273 = 303$ K
Maximum pressure, $p_3 = p_4 = 60$ bar
Cut-off $= 4\%$ of stroke volume
Number of working cycles/sec. $= 3$.

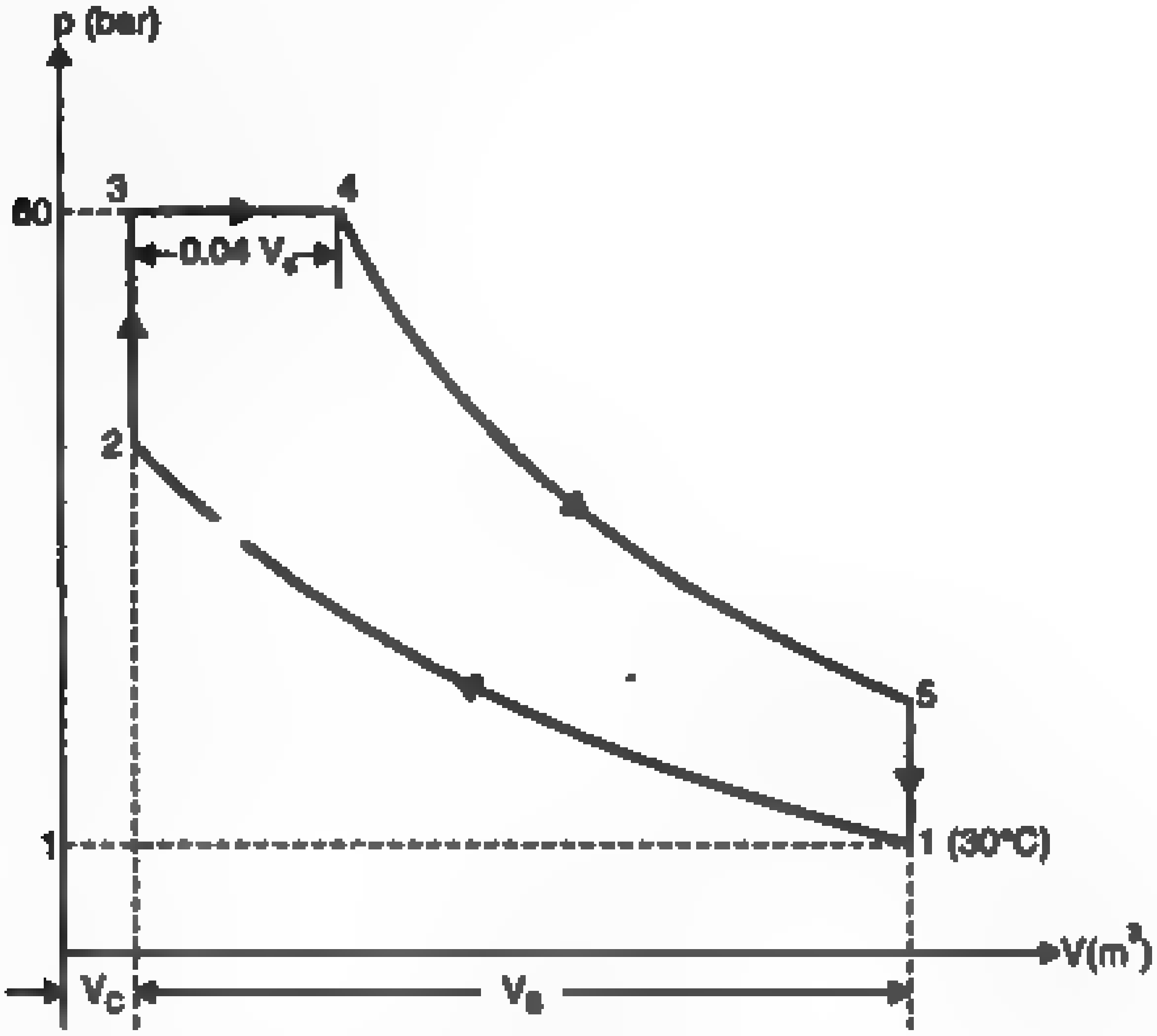


Fig. 3.21

(*i*) **Air standard efficiency :**

Now, swept volume, $V_s = \pi/4\ D^2L = \pi/4 \times 0.25^2 \times 0.3$
$= 0.0147\ m^3$

Also, compression ratio, $r = \dfrac{V_s + V_c}{V_c}$

i.e., $$9 = \frac{0.0147 + V_c}{V_c}$$

$\therefore$ $$V_c = \frac{0.0147}{8} = 0.0018\ m^3$$

$\therefore$ $$V_1 = V_s + V_c = 0.0147 + 0.0018 = 0.0165\ m^3$$

For the *adiabatic (or isentropic) process 1-2*,

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

$$p_2 = p_1 \times \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times (r)^{\gamma} = 1 \times (9)^{1.4} = 21.67 \text{ bar}$$

Also, $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (9)^{1.4-1} = (9)^{0.4} = 2.408$$

$\therefore \quad T_2 = T_1 \times 2.408 = 303 \times 2.408 = 729.6 \text{ K}$

For the *constant volume process 2-3,*

$$\frac{T_3}{p_3} = \frac{T_2}{p_2}$$

$$\therefore \quad T_3 = T_2 \cdot \frac{p_3}{p_2} = 729.6 \times \frac{60}{21.67} = 2020 \text{ K}$$

Also,
$$\frac{\rho - 1}{r - 1} = \frac{4}{100} \quad \text{or} \quad 0.04$$

$$\therefore \quad \frac{\rho - 1}{9 - 1} = 0.04 \quad \text{or} \quad \rho = 1.32$$

For the *constant pressure process 3-4,*

$$\frac{V_4}{T_4} = \frac{V_3}{T_3} \quad \text{or} \quad \frac{T_4}{T_3} = \frac{V_4}{V_3} = \rho$$

$$\therefore \quad T_4 = T_3 \times \rho = 2020 \times 1.32 = 2666.4 \text{ K}$$

Also expansion ratio,
$$\frac{V_5}{V_4} = \frac{V_5}{V_3} \times \frac{V_3}{V_4} = \frac{V_1}{V_2} \times \frac{V_2}{V_4} = \frac{r}{\rho} \qquad [\because V_5 = V_1 \text{ and } V_2 = V_3]$$

For *adiabatic process 4-5,*

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{\gamma - 1} = \left(\frac{\rho}{r}\right)^{\gamma - 1}$$

$$T_5 = T_4 \times \left(\frac{\rho}{r}\right)^{\gamma - 1} = 2666.4 \times \left(\frac{1.32}{9}\right)^{1.4 - 1} = 1237 \text{ K}$$

Also
$$p_4 V_4^{\gamma} = p_5 V_5^{\gamma}$$

$$p_5 = p_4 \cdot \left(\frac{V_4}{V_5}\right)^{\gamma} = 60 \times \left(\frac{r}{\rho}\right)^{\gamma} = 60 \times \left(\frac{1.32}{9}\right)^{1.4} = 4.08 \text{ bar}$$

Heat supplied,
$$Q_s = c_v(T_3 - T_2) + c_p(T_4 - T_3)$$
$$= 0.71\,(2020 - 729.6) + 1.0\,(2666.4 - 2020) = 1562.58 \text{ kJ/kg}$$

Heat rejected,
$$Q_r = c_v\,(T_5 - T_1)$$
$$= 0.71\,(1237 - 303) = 663.14 \text{ kJ/kg}$$

$$\eta_{\text{air-standard}} = \frac{Q_s - Q_r}{Q_s} = \frac{1562.85 - 663.14}{1562.58} = 0.5756 \text{ or } 57.56\%. \quad \textbf{(Ans.)}$$

(*ii*) Power developed by the engine, P :

Mass of air in the cycle is given by

$$m = \frac{p_1 V_1}{R T_1} = \frac{1 \times 10^5 \times 0.0165}{287 \times 303} = 0.0189 \text{ kg}$$

$\therefore$ Work done per cycle $= m(Q_s - Q_r)$

$= 0.0189\,(1562.58 - 663.14) = 16.999 \text{ kJ}$

Power developed $=$ Work done per cycle $\times$ No. of cycles per second

$= 16.999 \times 3 = 50.99$ say 51 kW. **(Ans.)**

Example 3.26. *In an engine working on Dual cycle, the temperature and pressure at the beginning of the cycle are 90°C and 1 bar respectively. The compression ratio is 9. The maximum pressure is limited to 68 bar and total heat supplied per kg of air is 1750 kJ. Determine :*

(*i*) *Pressure and temperatures at all salient points*

(*ii*) *Air standard efficiency*

(*iii*) *Mean effective pressure.*

Solution. Refer Fig. 3.22.

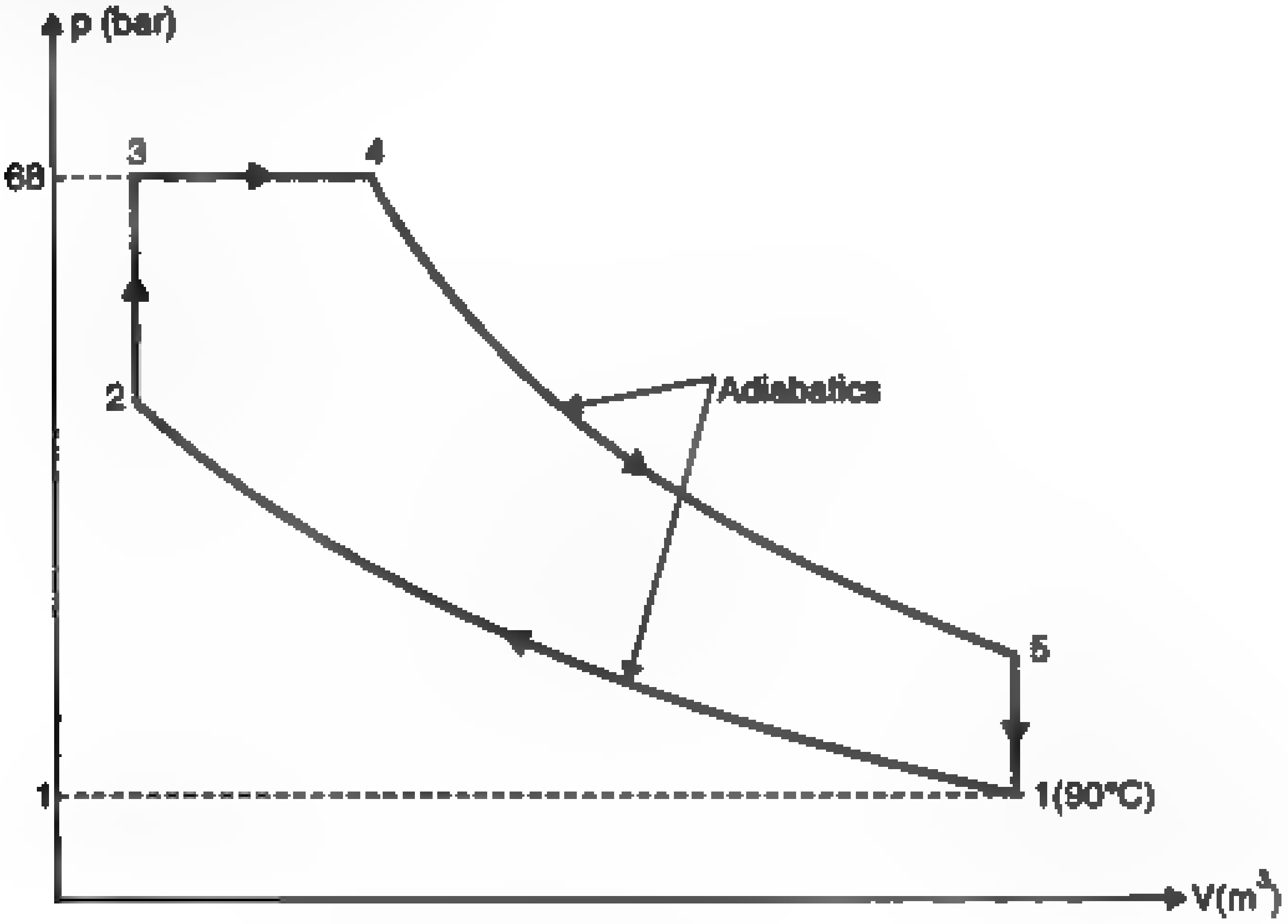


Fig. 3.22

Initial pressure, $p_1 = 1$ bar

Initial temperature, $T_1 = 90 + 273 = 363$ K

Compression ratio, $r = 9$

Maximum pressure, $p_3 = p_4 = 68$ bar

Total heat supplied $= 1750$ kJ/kg

(*i*) **Pressures and temperatures at salient points :**

For the *isentropic process* 1-2,

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

$$\mathbf{p_2} = p_1 \times \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times (r)^{\gamma} = 1 \times (9)^{1.4} = \mathbf{21.67\ bar.} \quad \textbf{(Ans.)}$$

Also, $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} = (9)^{1.4 - 1} = 2.408$$

$\therefore$ $$\mathbf{T_2} = T_1 \times 2.408 = 363 \times 2.408 = \mathbf{874.1\ K.} \quad \textbf{(Ans.)}$$

$$\mathbf{p_3 = p_4 = 68\ bar.} \quad \textbf{(Ans.)}$$

For the *constant volume process* 2-3,

$$\frac{p_2}{T_2} = \frac{p_3}{T_3}$$

$\therefore$ $$\mathbf{T_3} = T_2 \times \frac{p_3}{p_2} = 874.1 \times \frac{68}{21.67} = \mathbf{2742.9\ K.} \quad \textbf{(Ans.)}$$

Heat added at constant volume

$$= c_v (T_3 - T_2) = 0.71 (2742.9 - 874.1) = 1326.8 \text{ kJ/kg}$$

$\therefore$ Heat added at constant pressure

$$= \text{Total heat added} - \text{Heat added at constant volume}$$
$$= 1750 - 1326.8 = 423.2 \text{ kJ/kg}$$

$\therefore$ $$c_p(T_4 - T_3) = 423.2$$

or $$1.0(T_4 - 2742.9) = 423.2$$

$\therefore$ $$\mathbf{T_4} = \mathbf{3166\ K.} \quad \textbf{(Ans.)}$$

For *constant pressure process 3-4*,

$$\rho = \frac{V_4}{V_3} = \frac{T_4}{T_3} = \frac{3166}{2742.9} = 1.15$$

For *adiabatic (or isentropic) process 4-5*,

$$\frac{V_5}{V_4} = \frac{V_5}{V_3} \times \frac{V_3}{V_4} = \frac{V_1}{V_2} \times \frac{V_3}{V_4} = \frac{r}{\rho} \qquad \left(\because \ \rho = \frac{V_4}{V_3}\right)$$

Also $$p_4 V_4^{\gamma} = p_5 V_5^{\gamma}$$

$\therefore$ $$\mathbf{p_5} = p_4 \times \left(\frac{V_4}{V_5}\right)^{\gamma} = 68 \times \left(\frac{\rho}{r}\right)^{\gamma} = 68 \times \left(\frac{1.15}{9}\right)^{1.4} = \mathbf{3.81\ bar.} \quad \textbf{(Ans.)}$$

Again, $$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{\gamma-1} = \left(\frac{\rho}{r}\right)^{\gamma-1} = \left(\frac{1.15}{9}\right)^{1.4-1} = 0.439$$

$\therefore$ $$\mathbf{T_5} = T_4 \times 0.439 = 3166 \times 0.439 = \mathbf{1389.8\ K.} \quad \textbf{(Ans.)}$$

(*ii*) Air standard efficiency :

Heat rejected during constant volume process 5-1,

$$Q_r = C_v(T_5 - T_1) = 0.71(1389.8 - 363) = 729 \text{ kJ/kg}$$

$\therefore$ $$\eta_{\text{air-standard}} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{Q_s - Q_r}{Q_s}$$

$$= \frac{1750 - 729}{1750} = \mathbf{0.5834 \text{ or } 58.34\%.} \quad \textbf{(Ans.)}$$

(*iii*) Mean effective pressure, p_m :

Mean effective pressure is given by

$$p_m = \frac{\text{Work done per cycle}}{\text{Stroke volume}}$$

or $$p_m = \frac{1}{V_s}\left[p_3 (V_4 - V_3) + \frac{p_4 V_4 - p_5 V_5}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}\right]$$

$$V_1 = V_5 = r\,V_c,\ V_2 = V_3 = V_c,\ V_4 = \rho V_c,$$
$$V_s = (r-1)\,V_c \qquad \left[\begin{array}{l}\because \quad r = \dfrac{V_s + V_c}{V_c} = 1 + \dfrac{V_s}{V_c} \\ \therefore \ V_s = (r-1)\,V_c\end{array}\right]$$

$$\therefore \qquad p_m = \frac{1}{(r-1)V_c}\left[p_3(\rho V_c - V_c) + \frac{p_4\rho V_c - p_5 \times rV_c}{\gamma - 1} - \frac{p_2V_c - p_1rV_c}{\gamma - 1}\right]$$

$r = 9$, $\rho = 1.15$, $\gamma = 1.4$

$p_1 = 1$ bar, $p_2 = 21.67$ bar, $p_3 = p_4 = 68$ bar, $p_5 = 3.81$ bar

Substituting the above values in the above equation, we get

$$p_m = \frac{1}{(9-1)}\left[68(1.15-1) + \frac{68 \times 1.15 - 3.81 \times 9}{1.4-1} - \frac{21.67 - 9}{1.4-1}\right]$$

$$= \frac{1}{8}\,(10.2 + 109.77 - 31.67) = 11.04 \text{ bar}$$

Hence, *mean effective pressure* = **11.04 bar. (Ans.)**

Example 3.27. *An I.C. engine operating on the dual cycle (limited pressure cycle) the temperature of the working fluid (air) at the beginning of compression is 27°C. The ratio of the maximum and minimum pressures of the cycle is 70 and compression ratio is 15. The amounts of heat added at constant volume and at constant pressure are equal. Compute the air standard thermal efficiency of the cycle. State three main reasons why the actual thermal efficiency is different from the theoretical value.*

Take γ for air = 1.4. **(U.P.S.C. 1997)**

Solution. Refer Fig. 3.23. *Given :* $T_1 = 27 + 273 = 300$ K ; $\dfrac{p_3}{p_1} = 70$, $\dfrac{v_1}{v_2} = \dfrac{v_1}{v_3} = 15$

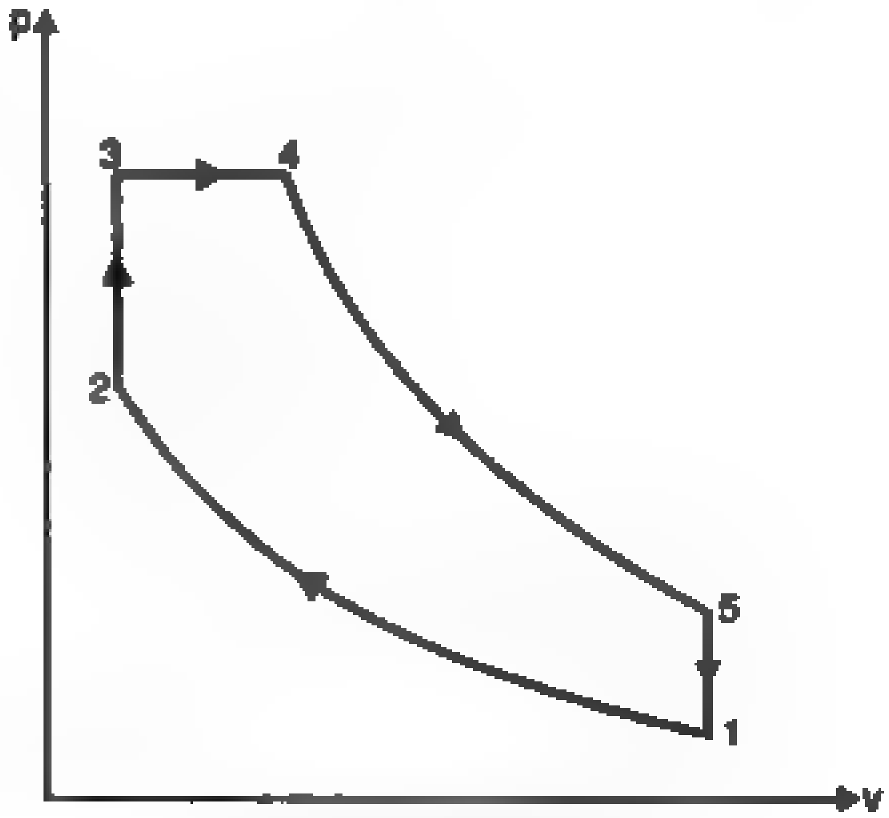


Fig. 3.23. Dual cycle.

Air standard efficiency, $\eta_{air\text{-}standard}$:

Consider *1 kg of air.*

Adiabatic compression process 1–2 :

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (15)^{1.4-1} = 2.954$$

3. Effect of variable specific heat, heat loss through cylinder walls, inlet and exhaust velocities of air/gas etc. have *not* been taken into account.

Example 3.28. *A Diesel engine working on a dual combustion cycle has a stroke volume of 0.0085 m^3 and a compression ratio 15 : 1. The fuel has a calorific value of 43890 kJ/kg. At the end of suction, the air is at 1 bar and 100°C. The maximum pressure in the cycle is 65 bar and air fuel ratio is 21 : 1. Find for ideal cycle the thermal efficiency. Assume c_p = 1.0 kJ/kg K and c_v = 0.71 kJ/kg K.*

Solution. Refer Fig. 3.24.

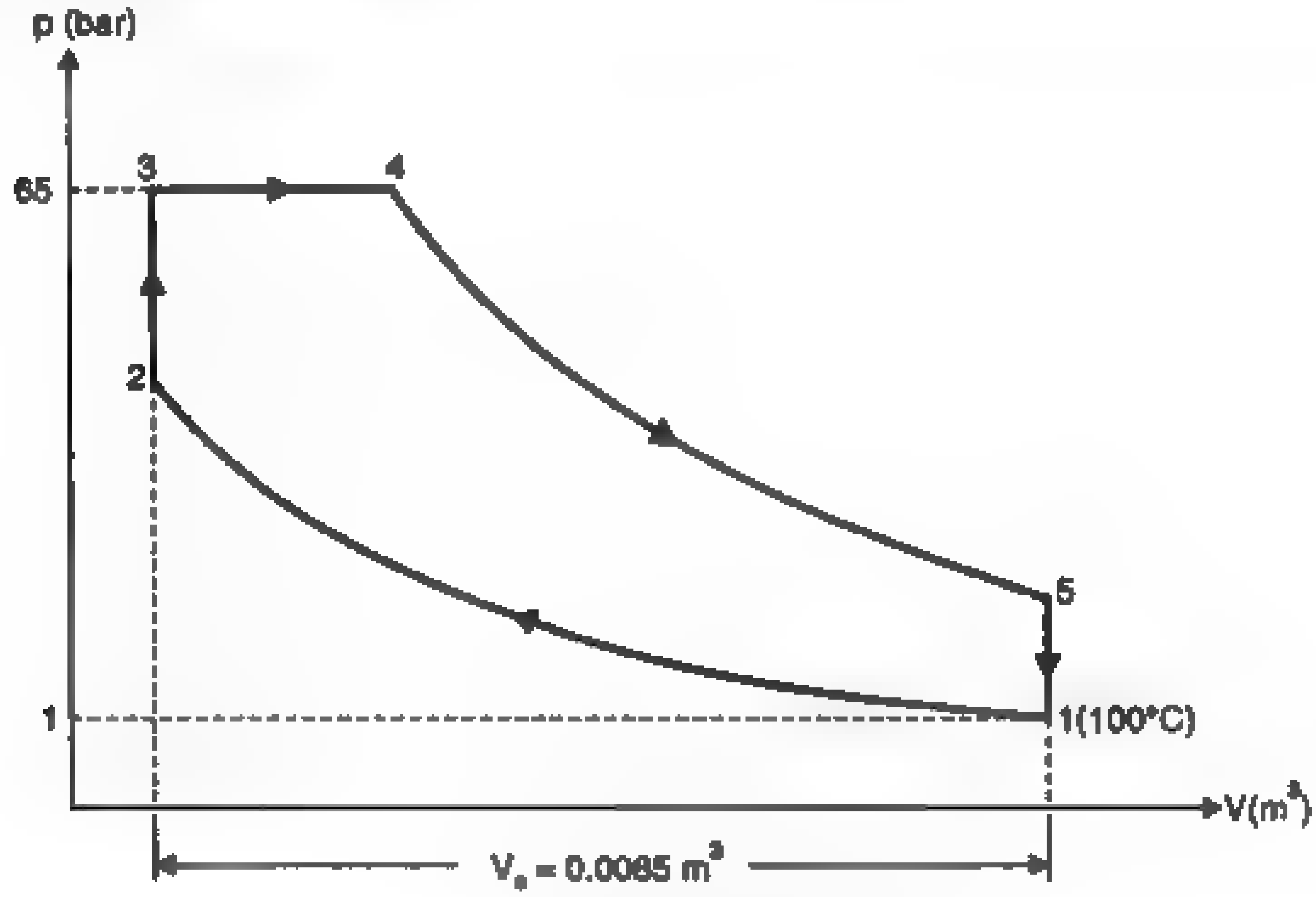


Fig. 3.24

Initial temperature, $T_1 = 100 + 273 = 373$ K

Initial pressure, $p_1 = 1$ bar

Maximum pressure in the cycle, $p_3 = p_4 = 65$ bar

Stroke volume, $V_s = 0.0085\ m^3$

Air-fuel ratio $= 21 : 1$

Compression ratio, $r = 15 : 1$

Calorific value of fuel, $C = 43890$ kJ/kg

$c_p = 1.0$ kJ/kg K, $c_v = 0.71$ kJ/kg K

Thermal efficiency :

$$V_s = V_1 - V_2 = 0.0085\ m^3$$

and as $$r = \frac{V_1}{V_2} = 15, \text{ then } V_1 = 15V_2$$

$\therefore$ $$15V_2 - V_2 = 0.0085$$

or $$14V_2 = 0.0085$$

or $$V_2 = V_3 = V_c = \frac{0.0085}{14} = 0.0006\ m^3$$

For *adiabatic compression process 1-2,*

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

or
$$p_2 = p_1 \cdot \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times (15)^{1.41} \qquad \left[\gamma = \frac{c_p}{c_v} = \frac{1.0}{0.71} = 1.41\right]$$

$$= 45.5 \text{ bar}$$

Also,
$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} = (15)^{1.41 - 1} = 3.04$$

$\therefore$
$$T_2 = T_1 \times 3.04 = 373 \times 3.04 = 1134 \text{ K or } 861^\circ\text{C}$$

For *constant volume process 2-3,*

$$\frac{p_2}{T_2} = \frac{p_3}{T_3}$$

or
$$T_3 = T_2 \times \frac{p_3}{p_2} = 1134 \times \frac{65}{45.5} = 1620 \text{ K or } 1347^\circ\text{C}$$

According to characteristic equation of gas,

$$p_1 V_1 = mRT_1$$

$\therefore$
$$m = \frac{p_1 V_1}{RT_1} = \frac{1 \times 10^5 \times 0.009}{287 \times 373} = 0.0084 \text{ kg (air)}$$

Heat added during constant volume process 2-3,

$$= m \times c_v (T_3 - T_2)$$
$$= 0.0084 \times 0.71 (1620 - 1134)$$
$$= 2.898 \text{ kJ}$$

Amount of fuel added during the *constant volume process 2-3,*

$$= \frac{2.898}{43890} = 0.000066 \text{ kg}$$

Also as air-fuel ratio is 21 : 1.

$\therefore$ Total amount of fuel added $= \dfrac{0.0084}{21} = 0.0004$ kg

Quantity of fuel added during the process 3-4,

$$= 0.0004 - 0.000066 = 0.000334 \text{ kg}$$

$\therefore$ *Heat added during the constant pressure operation 3-4*

$$= 0.000334 \times 43890 = 14.66 \text{ kJ}$$

But $(0.0084 + 0.0004)\, c_p (T_4 - T_3) = 14.66$

or $0.0088 \times 1.0 (T_4 - 1620) = 14.66$

$\therefore$
$$T_4 = \frac{\text{[illegible]}}{0.0088} + 1620 = 3286 \text{ K or } 3013^\circ\text{C}$$

Again for operation 3-4,

$$\frac{V_3}{T_3} = \frac{V_4}{T_4} \quad \text{or} \quad V_4 = \frac{V_3 T_4}{T_3} = \frac{0.0006 \times 3286}{1620} = 0.001217 \text{ m}^3$$

For adiabatic expansion operation 4-5,

$$\frac{T_4}{T_5} = \left(\frac{V_5}{V_4}\right)^{\gamma - 1} = \left(\frac{0.009}{0.001217}\right)^{1.41 - 1} = 2.27$$

or $$T_5 = \frac{T_4}{2.27} = \frac{3286}{2.27} = 1447.5 \text{ K or } 1174.5°\text{C}$$

Heat rejected during constant volume process 5-1,

$$= m\, c_v\, (T_5 - T_1)$$
$$= (0.00854 + 0.0004) \times 0.71\, (1447.5 - 373) = 6.713 \text{ kJ}$$

Work done $= \text{Heat supplied} - \text{Heat rejected}$

$$= (2.898 + 14.66) - 6.713 = 10.845 \text{ kJ}$$

∴ *Thermal efficiency,*

$$\eta_{th} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{10.845}{(2.898 + 14.66)} = 0.6176 \text{ or } 61.76\%. \quad \textbf{(Ans.)}$$

Example 3.29. *The compression ratio and expansion ratio of an oil engine working on the dual cycle are 9 and 5 respectively. The initial pressure and temperature of the air are 1 bar and 30°C. The heat liberated at constant pressure is twice the heat liberated at constant volume. The expansion and compression follow the law $pV^{1.25}$ = constant. Determine :*

(i) Pressures and temperatures at all salient points.

(ii) Mean effective pressure of the cycle.

(iii) Efficiency of the cycle.

(iv) Power of the engine if working cycles per second are 8.

Assume : Cylinder bore = 250 mm and stroke length = 400 mm.

Solution. Refer Fig. 3.25.

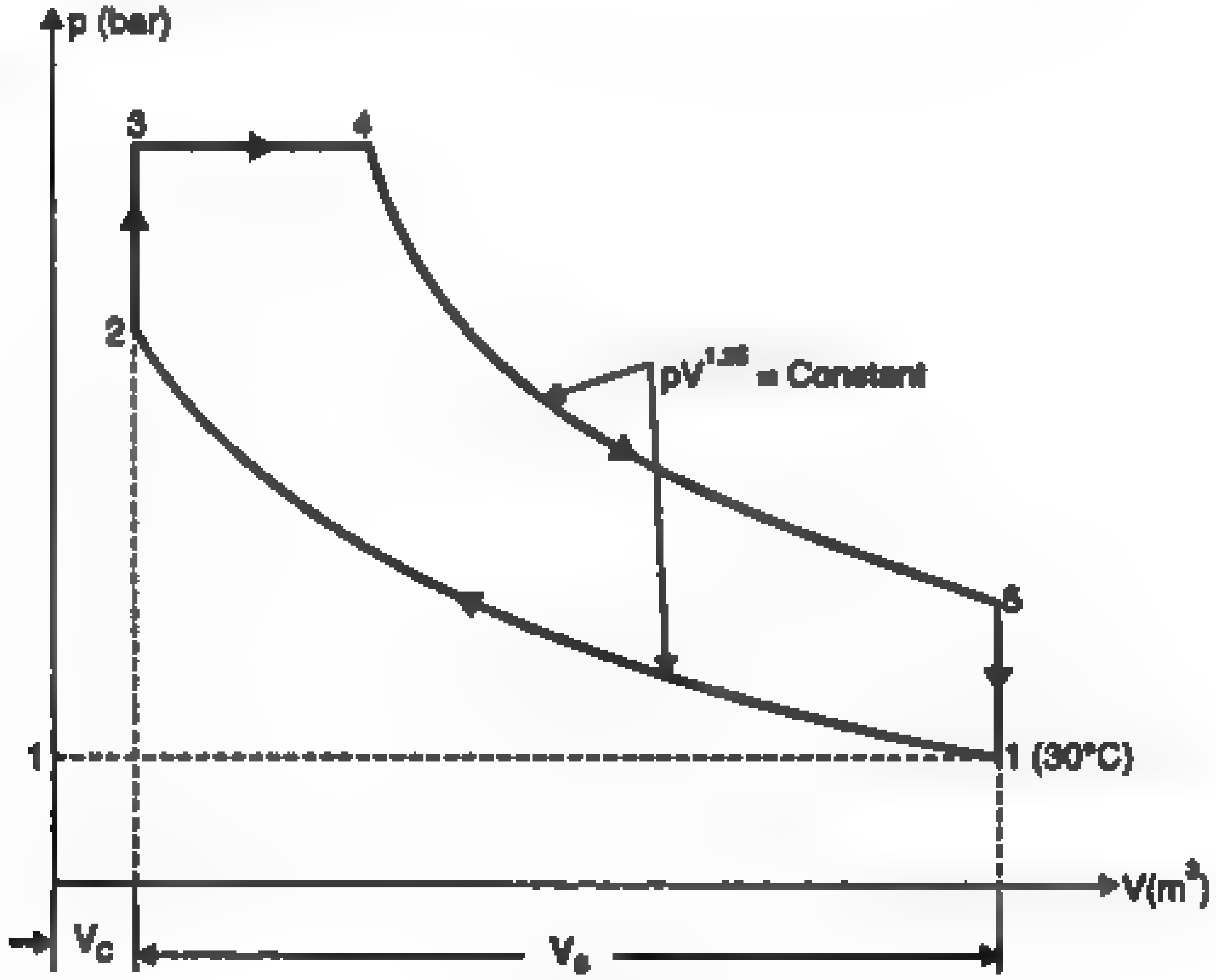


Fig. 3.25

Initial temperature, $T_1 = 30 + 273 = 303$ K

Initial pressure, $p_1 = 1$ bar

Compression and expansion law,

$$pV^{1.25} = \text{constant}$$

$$p_5 = p_4 \times \left(\frac{V_4}{V_5}\right)^n = p_4 \times \frac{1}{(r_e)^n} = \frac{35.7}{(5)^{1.25}} = 4.77 \text{ bar. (Ans.)}$$

Also $$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{n-1} = \frac{1}{(r_e)^{n-1}} = \frac{1}{(5)^{1.25-1}} = 0.668$$

$\therefore$ $$T_5 = T_4 \times 0.668 = 2163.4 \times 0.668 = 1445 \text{ K or } 1172°\text{C. (Ans.)}$$

(*ii*) **Mean effective pressure, p_m :**

Mean effective pressure is given by

$$p_m = \frac{1}{V_s}\left[p_3(V_4 - V_3) + \frac{p_4V_4 - p_5V_5}{n-1} - \frac{p_2V_2 - p_1V_1}{n-1}\right]$$

$$= \frac{1}{(r_c - 1)}\left[p_3(\rho - 1) + \frac{p_4\rho - p_5r_c}{n-1} - \frac{p_2 - p_1r_c}{n-1}\right]$$

Now, $r_c = \rho$, $\rho = 1.8$, $n = 1.25$, $p_1 = 1$ bar, $p_2 = 15.59$ bar, $p_3 = 35.7$ bar, $p_4 = 35.7$ bar, $p_5 = 4.77$ bar

$\therefore$ $$p_m = \frac{1}{(9-1)}\left[35.7(1.8-1) + \frac{35.7 \times 1.8 - 4.77 \times 9}{1.25 - 1} - \frac{15.59 - 1 \times 9}{1.25 - 1}\right]$$

$$= \frac{1}{8}[28.56 + 85.32 - 26.36] = 10.94 \text{ bar}$$

Hence *mean effective pressure* = **10.94 bar. (Ans.)**

(*iii*) **Efficiency of the cycle :**

Work done per cycle is given by $W = p_mV_s$

here $$V_s = \pi/4 D^2L = \pi/4 \times 0.25^2 \times 0.4 = 0.0196 \text{ m}^3$$

$\therefore$ $$W = \frac{10.94 \times 10^5 \times 0.0196}{1000} \text{ kJ/cycle} = 21.44 \text{ kJ/cycle}$$

Heat supplied per cycle $= mQ_s$

where m is the mass of air per cycle which is given by

$$m = \frac{p_1V_1}{RT_1} \quad \text{where } V_1 = V_s + V_c = \frac{r_c}{r_c - 1} V_s$$

$$\left[\begin{array}{l} r_c = \dfrac{V_s + V_c}{V_c} = 1 + \dfrac{V_s}{V_c} \quad \text{or} \quad V_c = \dfrac{V_s}{r_c - 1} \\ \therefore \; V_1 = V_s + \dfrac{V_s}{r_c - 1} = V_s\left(1 + \dfrac{1}{r_c - 1}\right) = \dfrac{r_c}{r_c - 1} V_s \end{array}\right]$$

$$= \frac{9}{9-1} \times 0.0196 = 0.02205 \text{ m}^3$$

$\therefore$ $$m = \frac{1 \times 10^5 \times 0.02205}{287 \times 303} = 0.02535 \text{ kg/cycle}$$

$\therefore$ Heat supplied per cycle

$$= mQ_s = 0.02535[c_v(T_3 - T_2) + c_p(T_4 - T_3)]$$
$$= 0.02535[0.71(1201.9 - 524.8) + 1.0(2163.4 - 1201.9)]$$
$$= 36.56 \text{ kJ/cycle}$$

$$\textit{Efficiency} = \frac{\text{Work done per cycle}}{\text{Heat supplied per cycle}} = \frac{21.44}{36.56}$$

$$= 0.5864 \quad \text{or} \quad 58.64\%. \text{ (Ans.)}$$

3.7.2. For the Same Compression Ratio and the Same Heat Input

A comparison of the cycles (Otto, Diesel and Dual) on the *p-v* and *T-s* diagrams for the *same compression ratio and heat supplied* is shown in the Fig. 3.27.

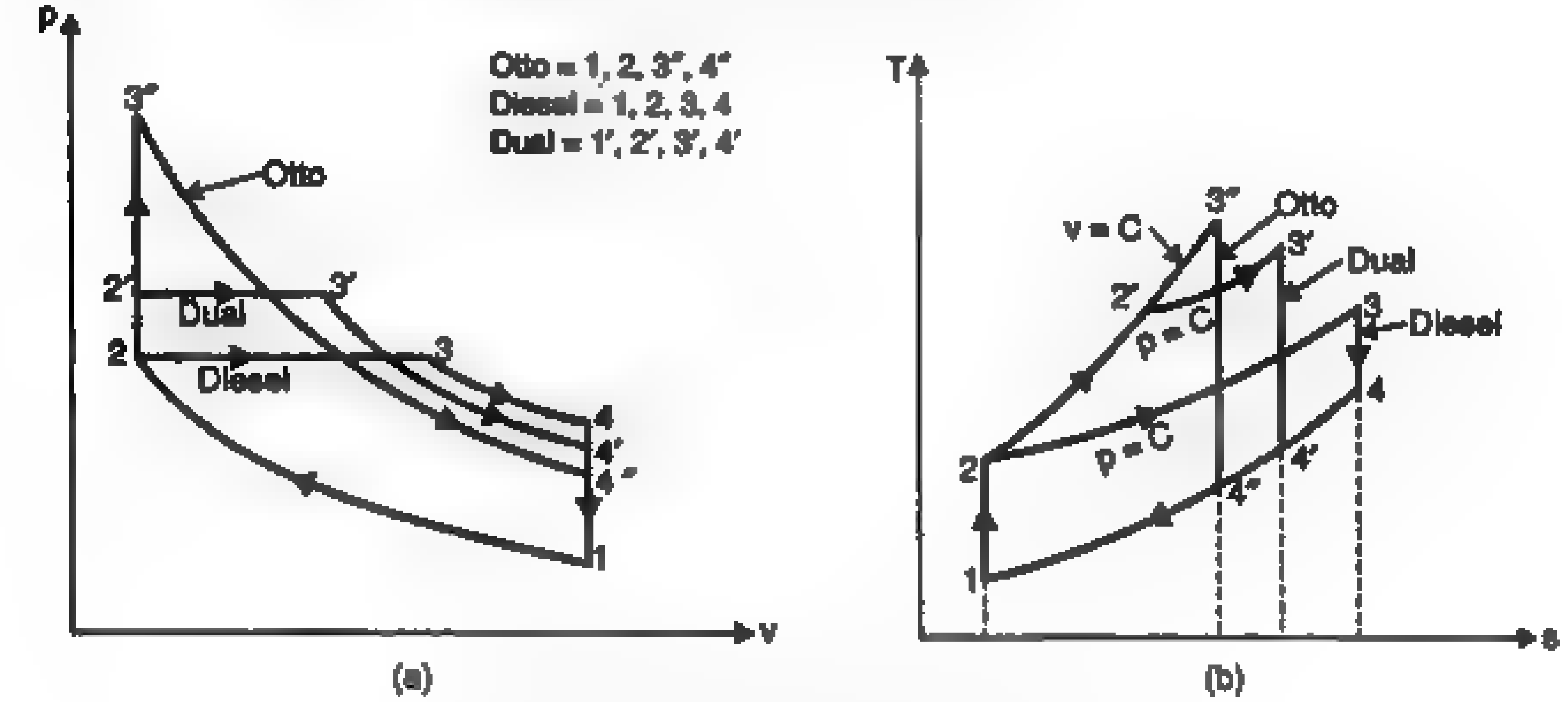


Fig. 3.27. (*a*) *p-v* diagram, (*b*) *T-s* diagram.

We know that, $$\eta = 1 - \frac{\text{Heat rejected}}{\text{Heat supplied}} \quad ...(3.13)$$

Since all the cycles reject their heat at the same specific volume, process line from state 4 to 1, the quantity *of heat rejected from each cycle is represented by the appropriate area under the line 4 to 1 on the T-s diagram.* As is evident from the eqn. (3.13) the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and Diesel cycle is the least efficient of the three cycles.

i.e., $$\eta_{otto} > \eta_{dual} > \eta_{diesel}.$$

3.7.3. For Constant Maximum Pressure and Heat Supplied

Fig. 3.28 shows the Otto and Diesel cycles on *p-v* and *T-s* diagrams for constant maximum pressure and heat input respectively.

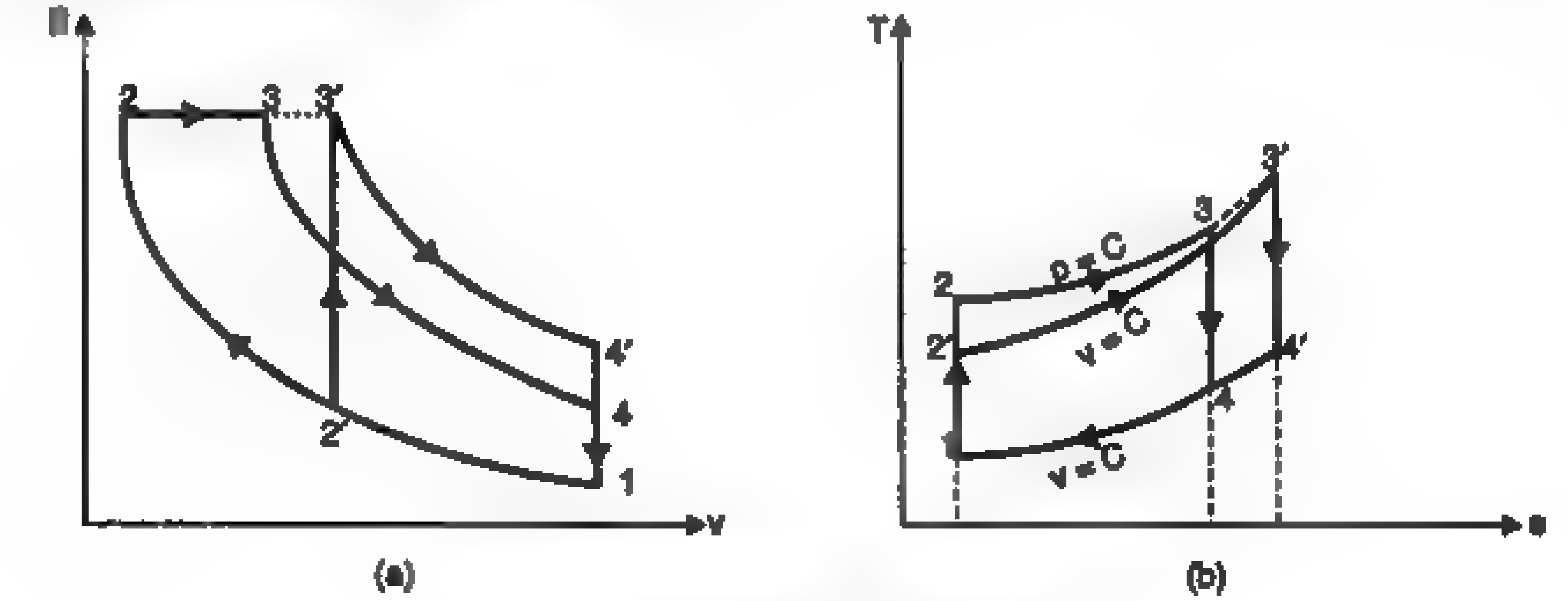


Fig. 3.28. (*a*) *p-v* diagram, (*b*) *T-s* diagram.

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(*i*) 1-2—*Heat rejection at constant pressure*

(*ii*) 2-3—*Adiabatic compression*

(*iii*) 3-4—*Addition of heat at constant volume*

(*iv*) 4-1—*Adiabatic expansion.*

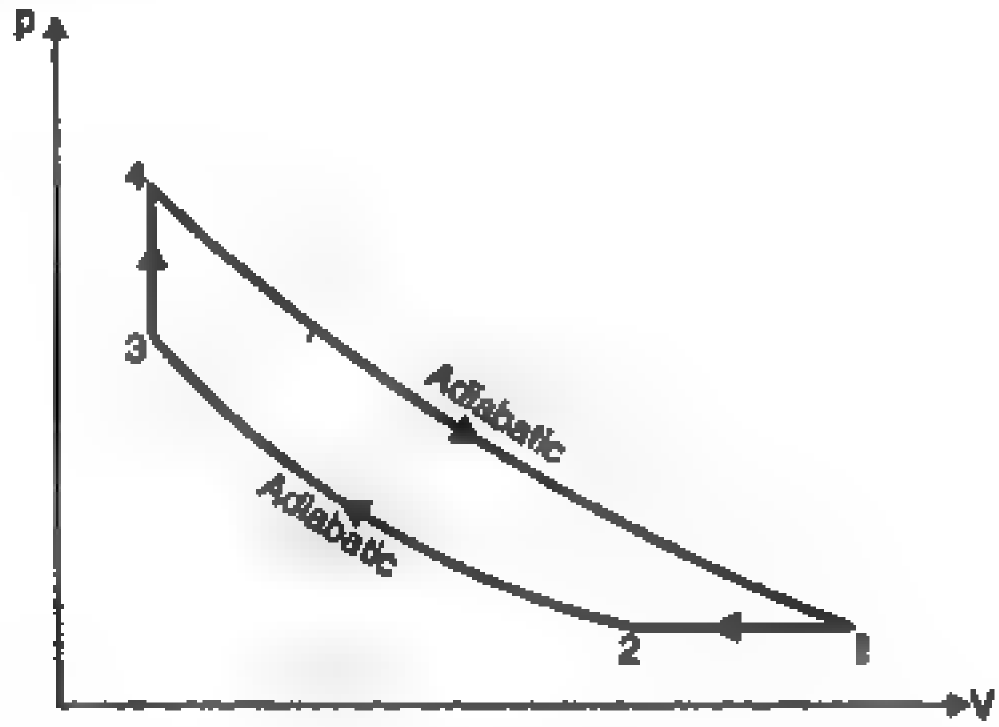

Fig. 3.30

Consider *1 kg of air*

Compression ratio $= \dfrac{v_2}{v_3} = \alpha$

Expansion ratio $= \dfrac{v_1}{v_4} = r$

Heat supplied at constant volume $= c_v(T_4 - T_3)$

Heat rejected $= c_p(T_1 - T_2)$

Work done = Heat supplied – Heat rejected

$= c_v(T_4 - T_3) - c_p(T_1 - T_2)$

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_4 - T_3) - c_p(T_1 - T_2)}{c_v(T_4 - T_3)}$$

$$= 1 - \gamma \cdot \frac{(T_1 - T_2)}{(T_4 - T_3)} \qquad \text{...(i)}$$

During *adiabatic compression 2-3,*

$$\frac{T_3}{T_2} = \left(\frac{v_2}{v_3}\right)^{\gamma - 1} = (\alpha)^{\gamma - 1}$$

or $$T_3 = T_2\,(\alpha)^{\gamma - 1} \qquad \text{...(ii)}$$

During constant pressure operation 1-2

$$\frac{v_1}{T_1} = \frac{v_2}{T_2}$$

or $$\frac{T_2}{T_1} = \frac{v_2}{v_1} = \frac{\alpha}{r} \qquad ...(iii)$$

$$\left(\frac{v_2}{v_1} = \frac{v_2}{v_3} \times \frac{v_3}{v_1} = \frac{v_2}{v_3} \times \frac{v_4}{v_1} = \frac{\alpha}{r}\right)$$

During *adiabatic expansion 4-1,*

$$\frac{T_4}{T_1} = \left(\frac{v_1}{v_4}\right)^{\gamma - 1} = (r)^{\gamma - 1}$$

$$T_1 = \frac{T_4}{(r)^{\gamma - 1}} \qquad ...(iv)$$

Putting the value of T_1 in equation *(iii)*, we get

$$T_2 = \frac{T_4}{(r)^{\gamma - 1}} \cdot \frac{\alpha}{r}$$

$$= \frac{\alpha T_4}{r^{\gamma}} \qquad ...(v)$$

Substituting the value of T_2 in equation *(ii)*, we get

$$T_3 = \frac{\alpha T_4}{(r)^{\gamma}} (\alpha)^{\gamma - 1} = \left(\frac{\alpha}{r}\right)^{\gamma} \cdot T_4$$

Finally putting the values of T_1, T_2 and T_3 in equation *(i)*, we get

$$\eta = 1 - \gamma \left(\frac{\frac{T_4}{r^{\gamma - 1}} - \frac{\alpha T_4}{(r)^{\gamma}}}{T_4 - \left(\frac{\alpha}{r}\right)^{\gamma} . T_4} \right) = 1 - \gamma \left(\frac{r - \alpha}{r^{\gamma} - \alpha^{\gamma}}\right)$$

Hence, *air standard efficiency* $= 1 - \gamma \cdot \left(\dfrac{r - \alpha}{r^{\gamma} - \alpha^{\gamma}}\right)$ ...(3.14)

Example 3.31. *A perfect gas undergoes a cycle which consists of the following processes taken in order :*

(*a*) *Heat rejection at constant pressure.*

(*b*) *Adiabatic compression from 1 bar and 27°C to 4 bar.*

(*c*) *Heat addition at constant volume to a final pressure of 16 bar.*

(*d*) *Adiabatic expansion to 1 bar.*

Calculate : (*i*) *Work done/kg of gas.*

(*ii*) *Efficiency of the cycle.*

Take : $c_p = 0.92$ *kJ/kg K,* $c_v = 0.75$ *kJ/kg K.*

Solution. Refer Fig. 3.31.

Pressure, $p_2 = p_1 = 1$ bar

Temperature, $T_2 = 27 + 273 = 300$ K

Pressure after adiabatic compression, $p_3 = 4$ bar

Final pressure after heat addition, $p_4 = 16$ bar

For *adiabatic compression 2-3,*

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.22 - 1}{1.22}} = 1.284 \qquad \left[\gamma = \frac{c_p}{c_v} = \frac{0.92}{0.75} = 1.22\right]$$

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The various operations are as follows :

Operation 1-2. The air is compressed isentropically from the lower pressure p_1 to the upper pressure p_2, the temperature rising from T_1 to T_2. No heat flow occurs.

Operation 2-3. Heat flows into the system increasing the volume from V_2 to V_3 and temperature from T_2 to T_3 whilst the pressure remains constant at p_2. Heat received $= mc_p\,(T_3 - T_2)$.

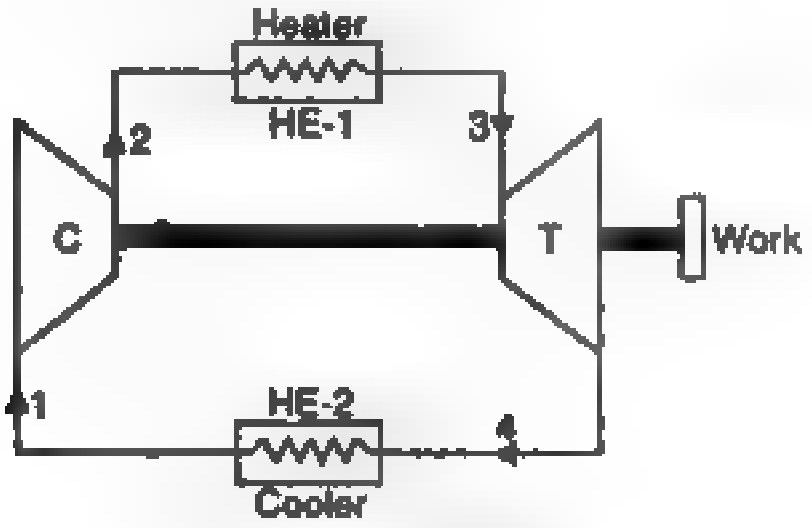
Operation 3-4. The air is expanded isentropically from p_2 to p_1, the temperature falling from T_3 to T_4. No heat flow occurs.

Operation 4-1. Heat is rejected from the system as the volume decreases from V_4 to V_1 and the temperature from T_4 to T_1 whilst the pressure remains constant at p_1. Heat rejected $= mc_p\,(T_4 - T_1)$.

$$\eta_{\text{air-standard}} = \frac{\text{Work done}}{\text{Heat received}}$$

$$= \frac{\text{Heat received/cycle} - \text{Heat rejected/cycle}}{\text{Heat received/cycle}}$$

$$= \frac{mc_p\,(T_3 - T_2) - mc_p\,(T_4 - T_1)}{mc_p\,(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$



C = Compressor T = Turbine
(a)

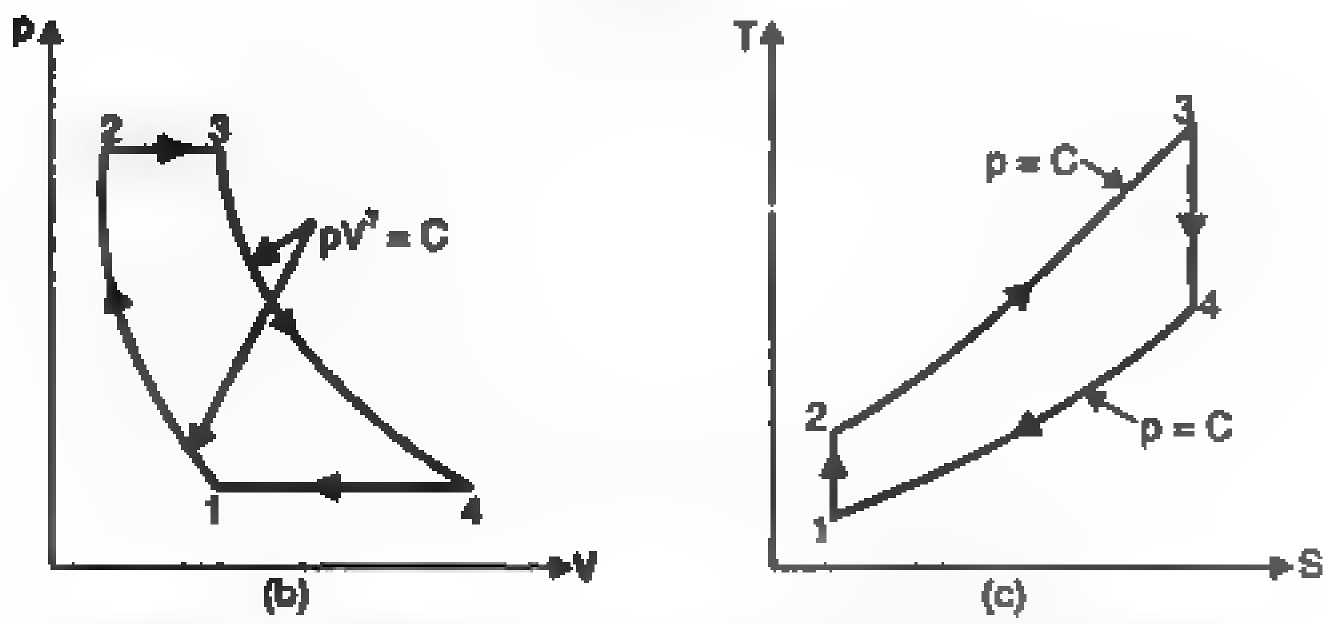


Fig. 3.33. Brayton cycle : (*a*) Basic components of a gas turbine power plant (*b*) *p-V* diagram (*c*) *T-S* diagram.

Now, from isentropic expansion

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = T_1\, (r_p)^{\frac{\gamma-1}{\gamma}} \text{ , where } r_p = \text{Pressure ratio.}$$

Similarly
$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{or} \quad T_3 = T_4\,(r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \qquad \eta_{\text{air-standard}} = 1 - \frac{T_4 - T_1}{T_4 (r_p)^{\frac{\gamma-1}{\gamma}} - T_1 (r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \qquad ...(3.16)$$

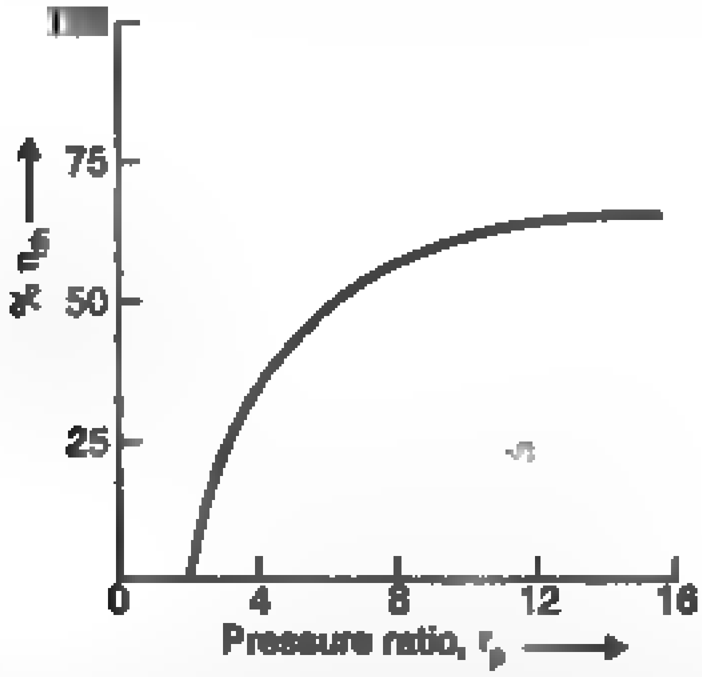


Fig. 3.34. Effect of pressure ratio on the efficiency of Brayton cycle.

The eqn. (3.16) shows that the *efficiency of the ideal joule cycle increases with the pressure ratio. The absolute limit of upper pressure is determined by the limiting temperature of the material of the turbine at the point at which this temperature is reached by the compression process alone, no further heating of the gas in the combustion chamber would be permissible and the work of expansion would ideally just balance the work of compression so that no excess work would be available for external use.*

Pressure ratio for maximum work :

Now we shall prove that the *pressure ratio for maximum work is a function of the limiting temperature ratio.*

Work output during the cycle

$$= \text{Heat received/cycle} - \text{Heat rejected/cycle}$$

$$= mc_p (T_3 - T_2) - mc_p (T_4 - T_1)$$

$$= mc_p (T_3 - T_4) - mc_p (T_2 - T_1)$$

$$= mc_p \left[T_3 \left(1 - \frac{T_4}{T_3}\right) - T_1 \left(\frac{T_2}{T_1} - 1\right) \right]$$

Solution. Pressure of intake air, $p_1 = 101\,325$ kPa

Temperature of intake air, $T_1 = 27 + 273 = 300$ K

The pressure ratio in the cycle, $r_p = 6$

(*i*) **Maximum temperature in the cycle, T_3 :**

Refer Fig 3.35.

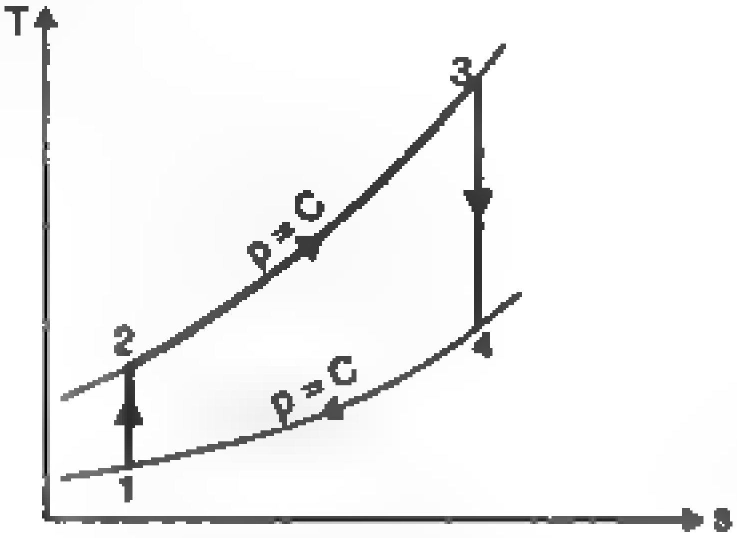


Fig 3.35

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.668$$

$$T_2 = 1.668\, T_1 = 1.668 \times 300 = 500.4 \text{ K}$$

Also, $$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.668$$

$$T_4 = \frac{T_3}{1.668}$$

But $$W_T = 2.5\, W_C \qquad \text{(Given)}$$

$\therefore$ $$mc_p\,(T_3 - T_4) = 2.5\, mc_p\,(T_2 - T_1)$$

or $$T_3 - \frac{T_3}{1.668} = 2.5\,(500.4 - 300) = 501 \text{ or } T_3\left(1 - \frac{1}{1.668}\right) = 501$$

$$\mathbf{T_3} = \frac{501}{\left(1 - \frac{1}{1.668}\right)} = 1251 \text{ K or } 978^\circ\text{C.} \quad \textbf{(Ans.)}$$

(*ii*) **Cycle efficiency, η_{cycle} :**

Now, $$T_4 = \frac{T_3}{1.668} = \frac{1251}{1.668} = 750 \text{ K}$$

$$\eta_{cycle} = \frac{\text{Net work}}{\text{Heat added}} = \frac{mc_p\,(T_3 - T_4) - mc_p\,(T_2 - T_1)}{mc_p(T_3 - T_2)}$$

$$= \frac{(1251 - 750) - (500.4 - 300)}{(1251 - 500.4)} = 0.4 \quad \text{or} \quad 40\%. \quad \textbf{(Ans.)}$$

$$\left[\ \textbf{Check}.\ \eta_{cycle} = 1 - \frac{1}{(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)}} = 1 - \frac{1}{(6)^{\frac{1.4-1}{1.4}}} = 0.4 \quad \text{or} \quad 40\%. \quad \textbf{(Ans.)}\ \right]$$

Example 3.33. *A gas turbine is supplied with gas at 5 bar and 1000 K and expands it adiabatially to 1 bar. The mean specific heat at constant pressure and constant volume are 1.0425 kJ/kg K and 0.7662 kJ/kg K respectively.*

(i) Draw the temperature-entropy diagram to represent the processes of the simple gas turbine system.

(ii) Calculate the power developed in kW per kg of gas per second and the exhaust gas temperature. **(GATE, 1995)**

Solution. *Given* : $p_1 = 1$ bar ; $p_2 = 5$ bar ; $T_3 = 1000$ K ; $c_p = 1.0425$ kJ/kg K ;
$c_v = 0.7662$ kJ/kg K.

$$\gamma = \frac{c_p}{c_v} = \frac{1.0425}{0.7662} = 1.36$$

(*i*) **Temperature-entropy diagram :**

Temperature-entropy diagram representing the processes of the simple gas turbine system is shown in Fig. 3.36.

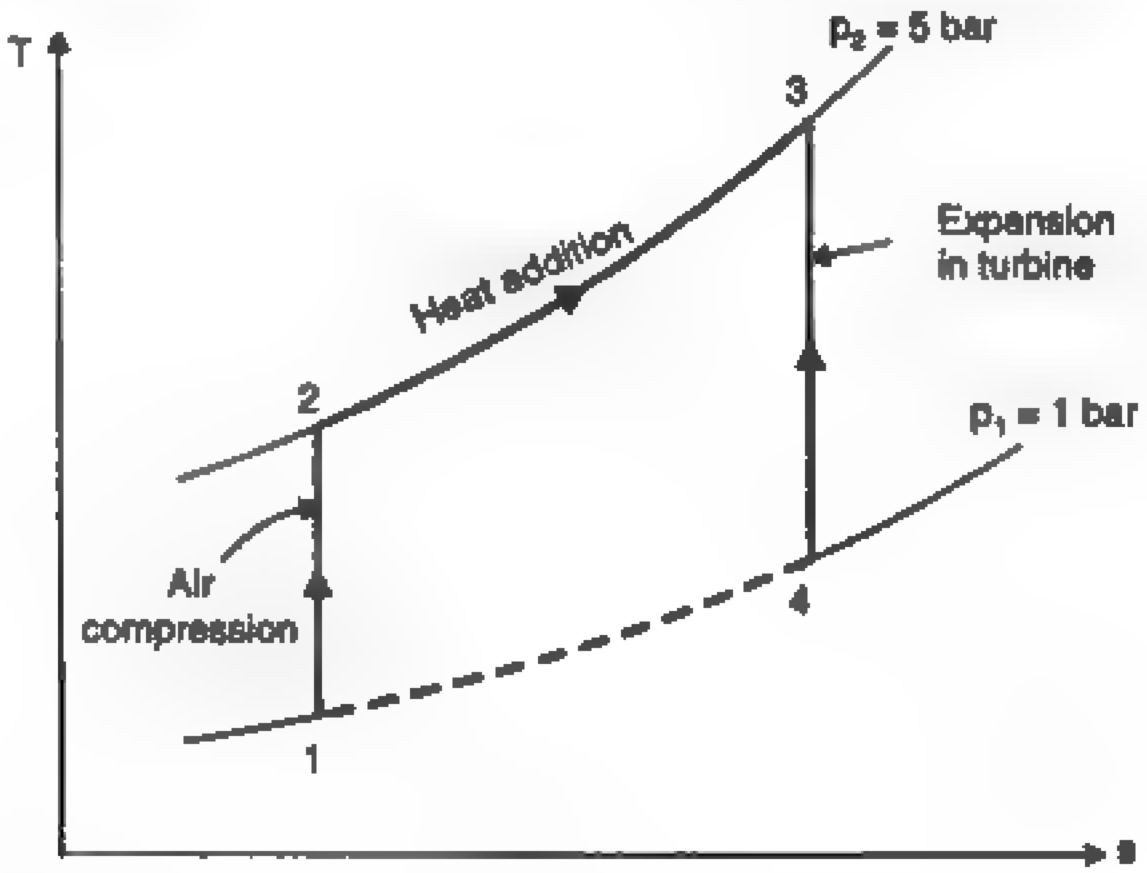


Fig. 3.36

(*ii*) **Power required :**

$$\frac{T_4}{T_3} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.36-1}{1.36}} = 0.653$$

$\therefore$ $T_4 = 1000 \times 0.653 = 653$ K

Power developed per kg of gas per second

$= c_p (T_3 - T_4)$

$= 1.0425 (1000 - 653) =$ **361.7 kW. (Ans.)**

Example 3.34. *An isentropic air turbine is used to supply 0.1 kg/s of air at 0.1 MN/m² and at 285 K to a cabin. The pressure at inlet to the turbine is 0.4 MN/m². Determine the temperature at turbine inlet and the power developed by the turbine. Assume c_p = 1.0 kJ/kg K.*

(GATE, 1999)

Example 3.38. *In a gas turbine plant working on Brayton cycle, the air at inlet is 27°C, 0.1 MPa. The pressure ratio is 6.25 and the maximum temperature is 800°C. The turbine and compressor efficiencies are each 80%. Find compressor work, turbine work, heat supplied, cycle efficiency and turbine exhaust temperature. Mass of air may be considered as 1 kg. Draw T-s diagram.* **(AMIE Summer, 2000)**

Solution. Refer Fig. 3.40.

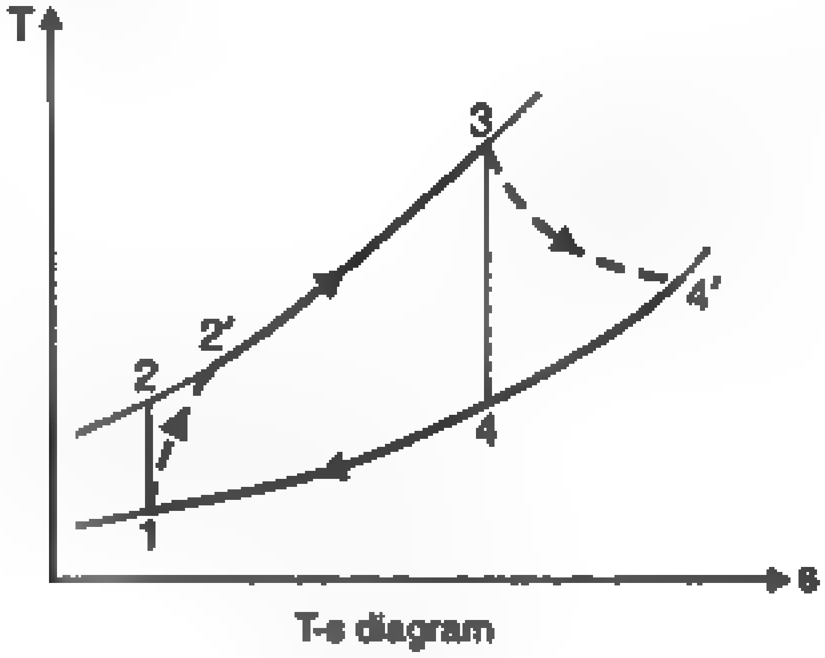


Fig. 3.40

Given : $T_1 = 27 + 273 = 300$ K ; $p_1 = 0.1$ MPa ; $r_p = 6.25$, $T_3 = 800 + 273 = 1073$ K

$\eta_{comp.} = \eta_{turbine} = 0.8$.

For the *compression process 1-2*, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6.25)^{\frac{1.4-1}{1.4}} = 1.688$$

or $$T_2 = 300 \times 1.688 = 506.4 \text{ K}$$

Also, $$\eta_{comp.} = \frac{T_2 - T_1}{T_2' - T_1} \quad \text{or} \quad 0.8 = \frac{506.4 - 300}{T_2' - 300}$$

or $$T_2' = \frac{506.4 - 300}{0.8} + 300 = 558 \text{ K}$$

$\therefore$ **Compressor work,** $W_{comp.} = 1 \times c_p \times (T_2' - T_1)$

$= 1 \times 1.005\ (558 - 300) =$ **259.29 kJ/kg. (Ans.)**

For *expansion process 3-4*, we have

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6.25)^{\frac{1.4-1}{1.4}} = 1.688$$

or $$T_4 = \frac{T_3}{1.688} = \frac{1073}{1.688} = 635.66 \text{ K}$$

Also, $$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} \quad \text{or} \quad 0.8 = \frac{1073 - T_4'}{1073 - 635.66}$$

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$= 287 \times 310 \ln (5.33) = 148878$ J/kg $= 148.88$ kJ/kg

Process 2-3 : $Q_{2\text{-}3} = W_{2\text{-}3} = 0$ (since volume is constant)

Process 3-4 : Heat supplied, $Q_{3\text{-}4} = W_{3\text{-}4} = RT_H \ln(r)$

$= 287 \times 930 \ln (5.33) = 446634$ J/kg or 446.63 kJ/kg

Process 4-1 : $Q_{4\text{-}1} = -c_v (T_L - T_H)$ or $c_v(T_H - T_L)$

Heat supplied during the process 2-3 = Heat rejected during the process 4-1.

$\therefore$ Work done = Net heat exchange during the isotherms

$= 446.63 - 148.88 = 297.75$ kJ/kg

$\therefore$ Thermal efficiency, $\eta_{th} = \dfrac{\text{Work done}}{\text{Heat supplied}} = \dfrac{297.75}{446.63} = 0.6676$ or **66.7%.**

Since Stirling cycle is *completely reversible*, its efficiency is also given as,

$$\eta = \frac{T_H - T_L}{T_H} = \frac{930 - 310}{930} = 0.667 \text{ or } \mathbf{66.7\%.} \quad \textbf{(Ans.)}$$

3.12. MILLER CYCLE

The Miller cycle (named after R. H. Miller) is a modern modification of the Atkinson cycle and has an *expansion ratio greater than the compression ratio*, which is accomplished, however, in much a different way. *Whereas a complicated mechanical linkage system of some kind is required for an engine designed to operate on the Atkinson cycle, a Miller cycle engine uses unique valve timing to obtain the same desired results.*

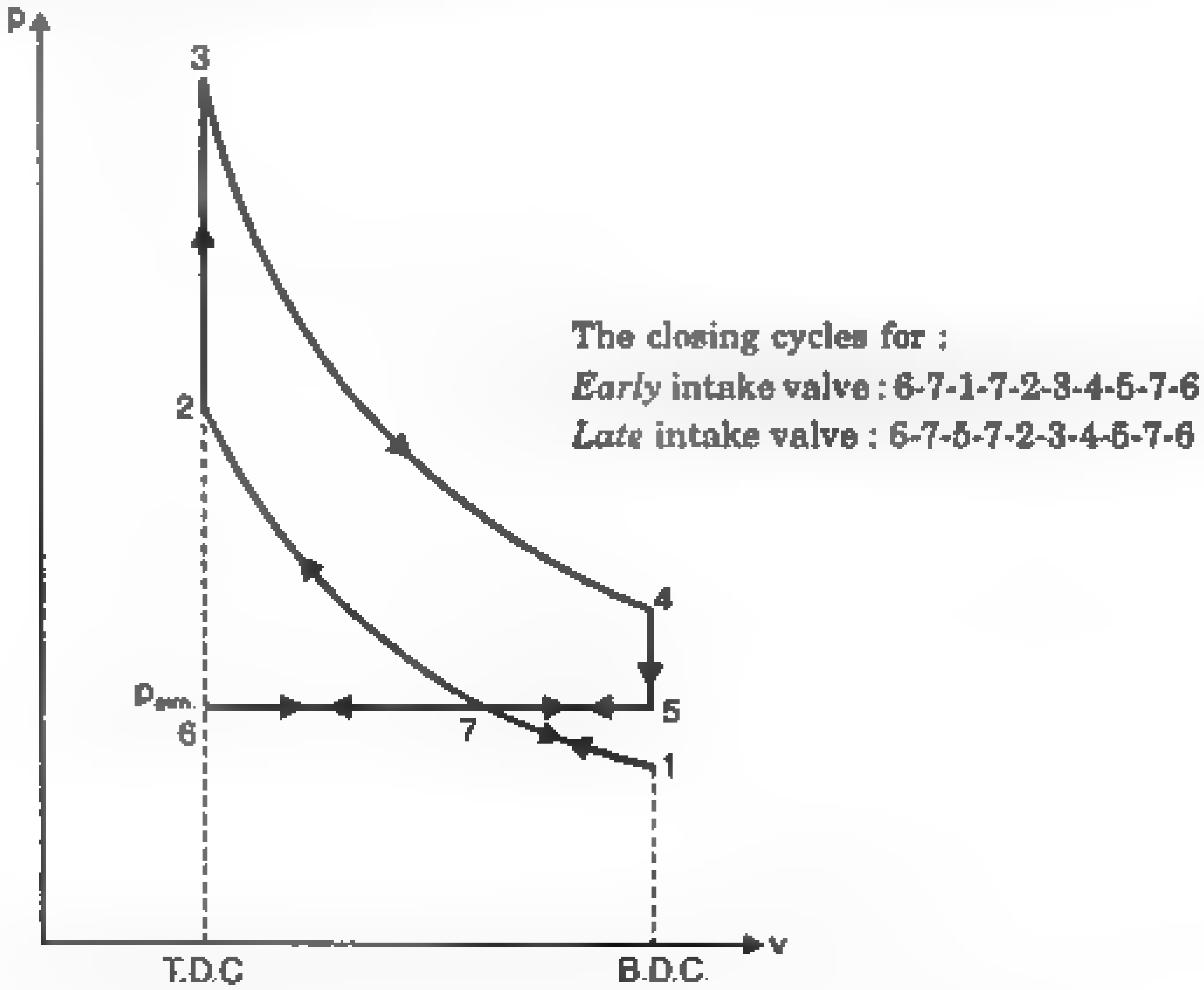

Fig. 3.43 Air-standard Miller cycle for unthrottled a naturally aspirated four-stroke cycle S.I. engine.

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2. The air standard efficiency of Otto cycle is given by

(a) $\eta = 1 + \frac{1}{(r)^{\gamma+1}}$ (b) $\eta = 1 - \frac{1}{(r)^{\gamma-1}}$

(c) $\eta = 1 - \frac{1}{(r)^{\gamma+1}}$ (d) $\eta = 2 - \frac{1}{(r)^{\gamma-1}}$.

3. The thermal efficiency of theoretical Otto cycle
   (a) increases with increase in compression ratio
   (b) increases with increase in isentropic index γ
   (c) does not depend upon the pressure ratio
   (d) follows all the above.
4. The work output of theoretical Otto cycle
   (a) increases with increase in compression ratio
   (b) increases with increase in pressure ratio
   (c) increases with increase in adiabatic index γ
   (d) follows all the above.
5. For same compression ratio
   (a) thermal efficiency of Otto cycle is greater than that of Diesel cycle
   (b) thermal efficiency of Otto cycle is less than that of Diesel cycle
   (c) thermal efficiency of Otto cycle is same as that for Diesel cycle
   (d) thermal efficiency of Otto cycle cannot be predicted.
6. In air standard Diesel cycle, at fixed compression ratio and fixed value of adiabatic index (γ)
   (a) thermal efficiency increases with increase in heat addition cut off ratio
   (b) thermal efficiency decreases with increase in heat addition cut off ratio
   (c) thermal efficiency remains same with increase in heat addition cut off ratio
   (d) none of the above.

ANSWERS

1. (b) 2. (b) 3. (d) 4. (d) 5. (a) 6. (b).

THEORETICAL QUESTIONS

1. What is a cycle ? What is the difference between an ideal and actual cycle ?
2. What is an air-standard efficiency ?
3. What is relative efficiency ?
4. Derive expressions of efficiency in the following cases :
   (i) Carnot cycle (ii) Diesel cycle (iii) Dual combustion cycle.
5. Explain "Air standard analysis" which has been adopted for I.C. engine cycles. State the assumptions made for air standard cycles.
6. Derive an expression for 'Atkinson cycle'.
7. Derive an expression for the thermal efficiency of Stirling cycle.
8. Explain the following cycles briefly and derive expressions of efficiency.
   (i) Miller cycle
   (ii) Lenoir cycle.

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4. The variation in the number of molecules present in the cylinder as the temperature and pressure change.

4.1.3. Assumptions made for Fuel-Air Cycle Analysis

Beside considering above factors, the following ***assumptions** are made for the analysis of fuel-air cycle :*

1. Prior to combustion there is no chemical change in either fuel or air.
2. Subsequent to combustion, the change is always in chemical equilibrium.
3. The processes are *adiabatic* (*i.e.,* there is no exchange of heat between the gases and cylinder walls in any process). In addition, the expansion and compression processes are *frictionless.*
4. The velocities are *negligibly small in case of reciprocating engines.*

Furthermore, in case of a *consant volume fuel-air cycle* the following assumptions are made :

- The fuel is completely vaporised and perfectly mixed with the air ;
- The burning takes place instantaneously at T.D.C. (at constant volume).

4.1.4. Importance of Fuel-Air Cycle

- Whereas the air standard cycle exhibits the general effect of compression ratio on efficiency of the engine, the fuel-air cycle may be calculated for various fuel-air ratios, inlet temperatures and pressures (It is worth noting that fuel-air ratio and compression ratio are much more important parameters in comparison to inlet conditions).
- With the help of fuel-air cycle analysis a *very good estimate of power to be expected from the actual engine can be made.* Furthermore, it is possible *to approximate very closely peak pressures and exhaust temperatures on which design and engine structure depend.*

4.1.5. Variable Specific Heats

4.1.5.1. General aspects

The specific heat of any substance is the ratio of the heat required to raise the temperature of a unit mass of the substance through one degree centigrade. Different substances have different values of specific heat. In case of gases, the temperature can be raised in two ways, *e.g.* either at *constant pressure* or *constant volume.* Accordingly we have two specific heats c_p and c_v. It is often convenient to use specific heats for the mol of a substance. A mol is M kilograms, called kilogram-mol abbreviated as kg mol. Here M is the molecular weight of the substance.

Thus *molar specific heat*

$$C = M.\ c \text{ kJ/mol K}$$

Similarly, $$C_p = M.\ c_p \text{ kJ/mol K}$$

and $$C_v = M.\ c_v \text{ kJ/mol K}$$

In general, the specific heats are *not constant. The specific heat varies largely with temperature but not very significantly with pressure except at very high pressure.* Thus in simple calculations, the variation in specific heat with pressure is neglected.

The specific heats of gases increase with the rise in temperature since the vibrational energy of the molecules increases with temperature. The effect of variable specific heats on the engine performance at higher temperature is considerable and it is, therefore, necessary to study these effects.

It is generally assumed that the specific heat is a linear function of temperature and the following relations hold good.

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$$= \frac{1}{(a-b)}\left[bp + \frac{bvdp}{dv} + KTp + \frac{KTvdp}{dv} + ap - bp\right]$$

$$= \frac{b}{(a-b)}\left[v.\frac{dp}{dv} + \frac{KTp}{b} + \frac{KTv}{b}.\frac{dp}{dv} + \frac{a}{b}p\right]$$

$$= \frac{1}{\left(\frac{a}{b}-1\right)}\left(\frac{a}{b}p + v.\frac{dp}{dv}\right) + \frac{KT}{(a-b)}\left(p + v.\frac{dp}{dv}\right)$$

$$\frac{a}{b} = \frac{c_p - KT}{c_v - KT} = \gamma'$$

$$\therefore \quad \frac{dQ}{dv} = \frac{1}{(\gamma'-1)}\left(\gamma' p + v.\frac{dp}{dv}\right) + \frac{KT}{(a-b)}\left(p + v.\frac{dp}{dv}\right) \quad ...(4.9)$$

If the expansion or compression is polytropic, then

pv^n = constant, where $n < \gamma'$.

Differentiating the above equation,

$$p \,.\, nv^{n-1} \,.\, dv + v^n \,.\, dp = 0$$

$$v \,.\, \frac{dp}{dv} = -p \,.\, n$$

Inserting this value in eqn. (4.9), the heat exchange in a polytropic process is given by

$$\frac{dQ}{dv} = \frac{1}{(\gamma'-1)}(\gamma' p - p \,.\, n) + \frac{KT}{(a-b)}(p - p \,.\, n)$$

$$= \frac{(\gamma'-n)}{(\gamma-1)} \,.\, p + \frac{KT}{(a-b)}(1-n)\,p = \left[\frac{\gamma'-n}{\gamma'-1} - \frac{(n-1)}{(a-b)}KT\right]p$$

$$\therefore \quad dQ = \left[\frac{(\gamma'-n)}{(\gamma'-1)} - \frac{(n-1)}{(a-b)}KT\right]pdv$$

$$= \left[\left(\frac{\gamma'-n}{\gamma'-1}\right) - \left(\frac{n-1}{a-b}\right)KT\right]dW \quad ...(4.10)$$

If $\quad K = 0$

$$dQ = \frac{\gamma'-n}{\gamma'-1} \,.\, dW.$$

4.1.5.4. Isentropic expansion with variable specific heats

The heat transfer to a system is expressed as :

$$dQ = du + dW$$

or $\quad dQ = c_v\, dT + pdv$ (considering one kg of air)

For isentropic process, $\quad dQ = 0$

$$\therefore \quad c_v\, dT + pdv = 0 \text{ or } c_v \frac{dT}{T} + \frac{pdv}{T} = 0$$

- The *reversible adiabatic expansion* from 3′ would be 3′-4″, but the expansion taking variable specific heat into account is above 3′-4″ and is represented by 3′-4′. The point 4′ will be below point 4 (the ideal expansion process starting from point 3 being along 3-4).

Thus, it is seen that the *effect of variation of specific heats is to lower temperatures and pressures at point 2 and 3* and *hence to deliver less work than the corresponding cycle with constant specific heats.*

4.1.7. Dissociation

- **Dissociation (*or chemical equilibrium loss*)** *refers to disintegration of burnt gases at high temperature. It is a reversible process and increases with temperature.*
- During dissociation a considerable amount of heat is absorbed ; this heat will be liberated when the elements recombine as the temperature falls. Thus the general effect of dissociation is suppression of a part of the heat during the combustion period and liberation of it as expansion proceeds, a condition which is really identical with the effects produced by the change in specific heat. However, *the effect of dissociation is much smaller than that of change of specific heat.*
- *The dissociation, in general, lowers the temperature and, consequently, the pressures at the beginning of the stroke, this causes a loss of power and efficiency.*
- The dissociation is mainly of CO_2 into CO and O_2 ;

$$2CO + O_2 \rightleftharpoons 2CO_2 + \text{heat}$$

The dissociation of CO_2 commences at about 1000°C and at 1500°C it amounts to 1 per cent.

There is very little dissociation of H_2O ;

$$2H_2 + O_2 \rightleftharpoons 2H_2O + \text{heat}$$

- *Dissociation is more severe in the chemically correct mixture.* If the mixture is *weaker, it gives temperatures lower than those required for dissociation to take place* while if it is *richer*, during combustion it will *give out CO and O_2 both of which suppress the dissociation of CO_2.*

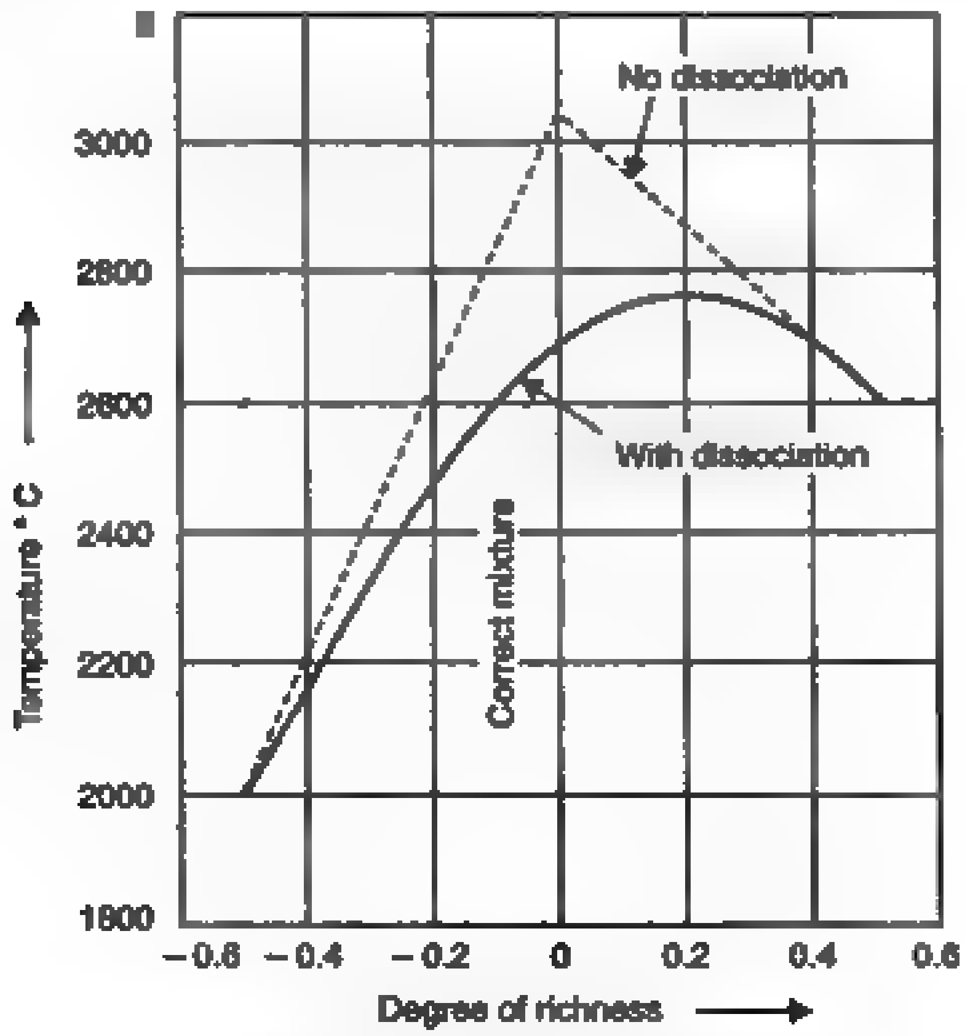

Fig. 4.4. Effect of dissociation on temperature at different mixture strength.

In view of the above, the air standard cycle concept for predicting the performance of S.I. engines is misleading, whereas the fuel-air cycle concept seems to be very reliable.

2. **Fuel-air ratio :**

(i) Efficiency

- It has been experimentally evaluated that the $\eta_{th,\,(I)}$ is highest at lean fuel-air mixtures of the order of $F_R = 0.85$ (As the mixture is made lean, due to less energy input the temperature rise during the combustion will be less which results in lower specific heat and eventually lower chemical equilibrium losses. This results in higher efficiency and as the fuel-air ratio is reduced, it approaches the air-cycle efficiency as illustrated in Fig. 4.10).
- As shown in Fig. 4.10, in the range of the mixture ratios of operation, for S.I. engines, usually $F_R = 0.6$ to 1.4, the fuel-air cycle very closely resembles the actual curve (experimental). The air-standard cycle concept miserably fails, *not influenced by the fuel-air ratio.*

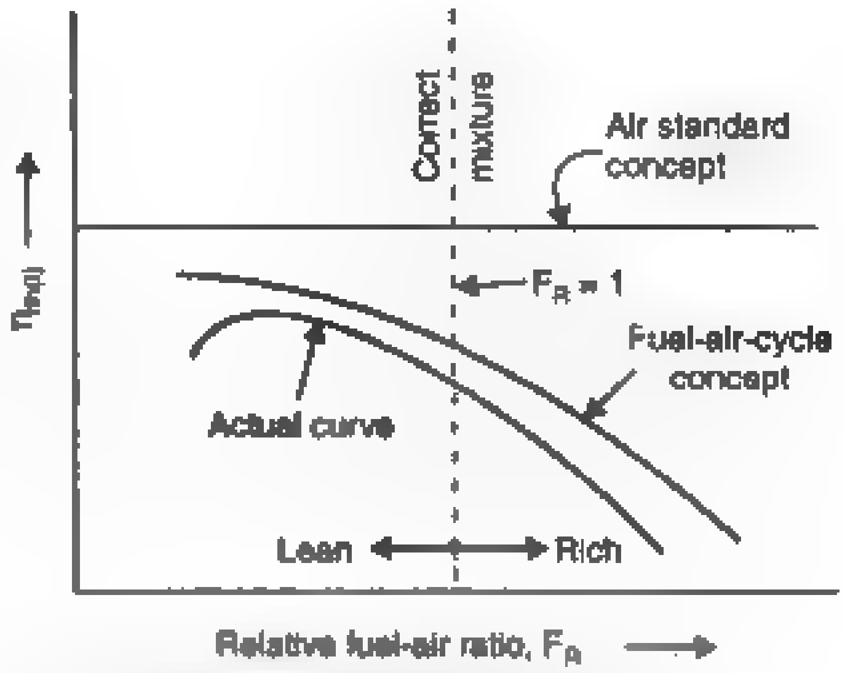

Fig. 4.10. Effect of mixture strength of $\eta_{th,(i)}$ at a given compression ratio.

(ii) Maximum power :

Fig. 4.11 shows the effect of mixture strength on cycle power.

- According to air-standard theory maximum power is at chemically correct mixture whereas by fuel-air theory maximum power is obtained when the mixture is about 10 per cent rich. The *efficiency drops rapidly as the mixture becomes enriched* ; this is due to the following *reasons* :
  - — Losses due to higher specific heats ;
  - — Chemical equilibrium losses ;
  - — Insufficient air which will result in formation of CO and H_2 in combustion, representing direct fuel wastage.

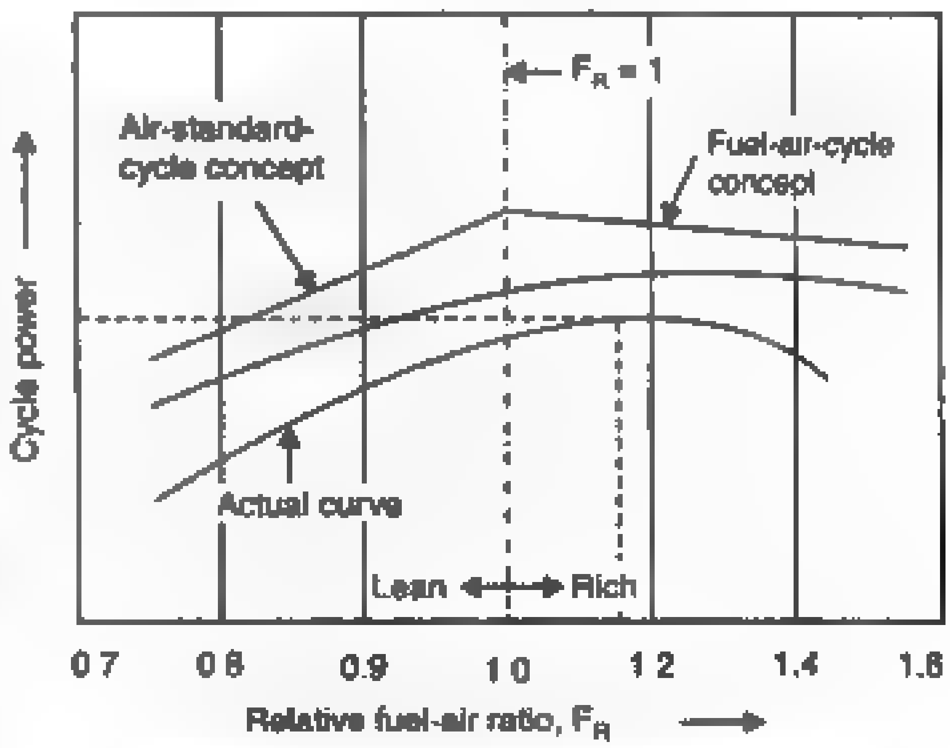

Fig. 4.11. Effect of mixture strength on cycle power

(iii) Maximum temperature :

Fig. 4.12 shows the effect of F_R on maximum cycle temperature $T_3(K)$ at different compression ratios.

— The maximum temperature at a given compression ratio is reached when the mixture is slightly rich, 6% or so as shown in Fig. 4.12.

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are constructed assuming enthalpy and internal energy to be zero at absolute zero temperature and integrating zero pressure specific heat equations from 0(K) to the given temperature T (K).

— The most important *assumption* is the validity of the equation $pv = RT$, where p, v, R and T denote pressure, specific volume, the characteristic gas constant and the absolute thermodynamic temperature respectively. This is *true when the gas has a critical temperature very low as compared to the range of temperature* met with in engineering applications. For air at 0°C and 20 bar the deviation is only 1 per cent ; and at 0°C and 1 bar, the deviation is 0.1 per cent.

— The enthalpy and internal energy are function of temperature and, therefore, their values can be computed with *single variable property, i.e., temperature.*

Thus enthalpy h and internal energy u at any temperature T (K), are given by .

$$h = \int_0^T c_p \, dT$$

$$u = \int_0^T c_v \, dT$$

The entropy change involves both variables namely pressure and temperature.

In gas tables, h, p_r, u, v_r and ϕ are recorded for different values of temperature T (K).

Relative pressure, p_r :

$$T\,ds = dh - v\,dp = c_p\,dT - v\,dp \qquad \ldots(4.20)$$

$$(\because \; dh = c_p dT)$$

For isentropic process, $ds = 0$, we get

$$0 = c_p\,dT - v\,dp \quad \text{or} \quad v\,dp = c_p\,dT$$

Dividing by $pv = RT$, we get

$$\frac{v\,dp}{pv} = \frac{c_p\,dT}{RT} \quad \text{or} \quad \frac{dp}{p} = \frac{c_p}{R}\,\frac{dT}{T}$$

or
$$\ln\left(\frac{p}{p_0}\right) = \frac{1}{R}\int_{T_0}^{T} c_p \,.\, \frac{dT}{T} = \ln(p_r) \qquad \ldots(4.21)$$

where T_0 is selected as a base temperature. It is seen that ratio $\frac{p}{p_0}$ is a function of temperature only, and is independent of the value of entropy.

— From eqn. (4.21) p_r can be calculated in terms of temperature.

On an isentropic path, for two states 1 and 2, we have

$$\frac{p_{r_1}}{p_{r_2}} = \frac{p_1/p_0}{p_2/p_0} = \frac{p_1}{p_2} \qquad \ldots(4.22)$$

Thus, the ratio of relative pressure for two states having the same entropy is equal to the ratio of the absolute pressures for the same two states ; p_0 is chosen as *unity* for computed values.

Relative volume, v_r :

$$T\,ds = du + pdv \qquad \ldots(4.23)$$

For an isentropic process, we have

$$0 = du + pdv$$

or
$$pdv = -\,du = -\,c_v dT$$

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Example 4.2. *Determine the effect of percentage change in the efficiency of Otto cycle having a compression ratio of 8, if the specific heat at constant volume increases by 1.1 percent.*

Solution. *Given* : Compression ratio, $r = 8$;

Increase in specific heat at constant volume, $\frac{dc_v}{c_v} = 1.1\%$

Percentage change in Otto cycle efficiency, $\frac{d\eta}{\eta}$:

The Otto cycle efficiency (η) is given by :

$$\eta = 1 - \frac{1}{(r)^{\gamma - 1}}$$

Now, $\quad c_p - c_v = R, \qquad \gamma - 1 = \frac{R}{c_v}$

$$\therefore \quad \eta = 1 - (r)^{-(\gamma-1)} = 1 - (r)^{-R/c_v}$$

or $$(1 - \eta) = (r)^{-R/c_v}$$

or $$\ln (1 - \eta) = -\frac{R}{c_v} \ln (r)$$

Differentiating both sides, we have

$$-\frac{1}{1-\eta} \cdot d\eta = \frac{R}{c_v^2} \ln (r) \times dc_v$$

or $$d\eta = -\frac{(1-\eta)R \ln (r)}{c_v^2} dc_v$$

or $$d\eta = -\frac{(1-\eta)(\gamma-1)c_v \times \ln (r)}{c_v^2} \cdot dc_v$$

$$\frac{d\eta}{\eta} = -\frac{(1-\eta)(\gamma-1) \ln (r)}{\eta} \cdot \frac{dc_v}{c_v}$$

Now $$\eta = 1 - \frac{1}{(8)^{1.4-1}} = 0.565$$

$$\therefore \quad \frac{d\eta}{\eta} = -\frac{(1-0.565)(1.4-1) \ln (8)}{0.565} \times \frac{1.1}{100}$$

$$= -0.704\% \text{ (decreased). (Ans.)}$$

Example 4.3. *Find the percentage change in efficiency of an Otto cycle for a compression ratio of 7 to 1 if the specific heat at constant volume increases by 3%.*

Solution. The change in efficiency with variation in specific heat is given by :

$$\frac{d\eta}{\eta} = -\frac{1-\eta}{\eta} (\gamma - 1) \log_e r \frac{dc_v}{c_v} \qquad ...(i)$$

Also, $$\eta = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(7)^{1.4-1}} = 0.541 \text{ or } 54.1\%$$

Mean specific heat during constant volume heat addition

$$c_{v(mean)} = 0.7117 + 2.1 \times 10^{-4} \left(\frac{T_3 + T_2}{2}\right)$$

Thus heat added at constant volume/kg of charge

$$= \frac{[redacted]}{16} = 3000 \text{ kJ/kg of charge}$$

And $$3000 = \left[0.7117 + 2.1 \times 10^{-4} \left(\frac{T_3 + 756}{2}\right)\right] \times (T_3 - 756)$$

$\therefore$ $$3000 = 0.7117\, T_3 + \frac{2.1 \times 10^{-4}}{2}\, T_3^2 + \frac{2.1 \times 10^{-4} \times 756}{2}\, T_3 - 0.7117 \times 756$$

$$- \frac{2.1 \times 10^{-4} \times 756}{2}\, T_3 - \frac{2.1 \times 10^{-4}}{2} \times 756 \times 756$$

$$= 0.7117\, T_3 + 0.000105\, T_3^2 - 598$$

$$= 0.000105\, T_3^2 + 0.7117\, T_3 - 598$$

or $$0.000105\, T_3^2 + 0.7117\, T_3 - 3598 = 0$$

Solving for T_3, $T_3 = 3370$ K

$\therefore$ *Maximum pressure in the cycle,*

$$p_3 = p_2 \times \frac{T_3}{T_2} = 22.91 \times \frac{3370}{756} = \mathbf{102.1 \text{ bar.}} \quad \textbf{(Ans.)}$$

When c_v remains constant at 0.7117 kJ/kg K ;

$$3000 = 0.7117\,(T_3 - 756)$$

or $$T_3 = \frac{3000}{0.7117} + 756 = 4971 \text{ K}$$

and $$p_3 = p_2\, \frac{T_3}{T_2} = 22.91 \times \frac{4971}{756} = \mathbf{150.6 \text{ bar.}} \quad \textbf{(Ans.)}$$

Example 4.7. *Combustion in a diesel engine is assumed to begin at inner dead centre and to be at constant pressure. The air-fuel ratio is 27 : 1, the calorific value of the fuel is 43000 kJ/kg, and the specific heat of the products of combustion is given by :*

$c_v = 0.71 + 20 \times 10^{-5}\, T$; R for the products = 0.287 kJ/kg K.

If the compression ratio is 15 : 1, and the temperature at the end of compression 870 K, find at what percentage of the stroke combustion is completed.

Solution. *Given* : Air-fuel ratio $= 27 : 1$

Calorific value of fuel, $c = 43000$ kJ/kg

Specific heat of product of combustion : $c_v = 0.71 + 20 \times 10^{-5}$ T

R for products $= 0.287$ kJ/kg K

Compression ratio, $r = 15 : 1$

Temperature at the end of compression, $T_2 = 870$ K

Percentage of the stroke when combustion is completed :

For 1 kg of fuel the charge is 28 kg and the heating value is 43000 kJ/kg

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The actual mixture strength (F/A ratio) expressed in terms of the chemically correct value is

$$\frac{15.37}{13.8} \times 100 = 111.4\%$$

i.e. the mixture used is 11.4% *rich in fuel*. The *combustion is, therefore, incomplete and hence CO will be formed.*

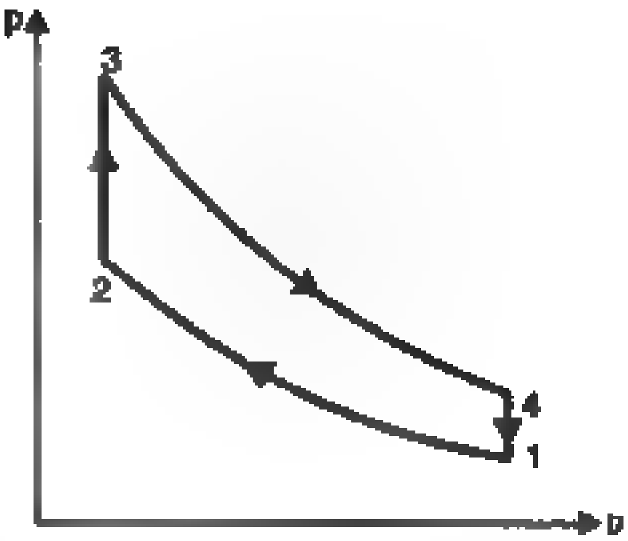

Fig. 4.20

$$1.114\ C_6H_{14} + 9.5O_2 = aCO_2 + bCO + cH_2O$$

Equating atoms of the same element before and after combustion, we get

$$1.114 \times 6 = a + b\ ;\ 1.114 \times 14 = 2c\ ;$$

$$9.5 \times 2 = 2a + b + c$$

or $\quad a + b = 6.684\ ;\ c = 7.798,\ 2a + b + c = 19$

Solving, we have : $\quad a = 4.52,\ b = 2.16,\ c = 7.8$

By adding nitrogen on both sides we get the actual combustion equation as given below :

$$1.114\ C_6H_{14} + 9.5\ O_2 + 9.5 \times (79/21)\ N_2 = 4.52\ CO_2 + 2.16\ CO + 7.8\ H_2O + 9.5 \times (79/21)\ N_2$$

$\therefore$ Moles before combustion $= 1.114 + 9.5 + 35.74 = 46.354$ say 46.35

Moles after combustion $= 4.52 + 2.16 + 7.8 + 35.74 = 50.22$

$$\left[\because\ \textit{Molecular expansion} = \frac{50.22 - 46.35}{46.35} = 0.0835 \text{ or } 8.35\%\right]$$

From *compression process 1-2*, we have :

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1}$$

$$\frac{T_2}{343} = (8)^{1.35-1} = 2.07 \qquad \therefore\ T_2 = 710 \text{ K}$$

Also $\quad Q_{2-3} = c_v(T_3 - T_2)$ for 1 kg of mixture

$$\frac{44 \times 10^3}{14.8} = 0.716\ (T_3 - 710) \quad \text{or} \quad T_3 = 4862 \text{ K}$$

Ignoring molecular expansion,

$$\frac{p_1v_1}{T_1} = \frac{p_3v_3}{T_3}$$

or
$$p_3 = p_1 \times \frac{v_1}{v_3} \times \frac{T_3}{T_1} = 1 \times 8 \times \frac{4862}{343} = 113.4 \text{ bar.} \quad \textbf{(Ans.)}$$

(*ii*) **Considering molecular contraction :**

Since mass of the reactants and products is same and specific heats are assumed same, the *temperature of the products with* molecular expansion will *remain same* as without molecular expansion ; *only the pressure will change*

$pv = nRT$, where n is the number of moles

$\therefore \quad p \propto n$

$\therefore$ Pressure with molecular expansion $= 113.4 \times \dfrac{[redacted]}{46.35} = 122.87$ bar. **(Ans.)**

13. Explain clearly the effect of compression ratio and mixture strength on thermal efficiency.
14. What is the effect of mixture strength on thermal efficiency at a given compression ratio.
15. What is the effect of mixture strength on cycle power ?
16. State the effect of F_R on maximum cycle temperature and pressure at different compression ratios
17. State the characteristics of constant volume fuel-air cycle.
18. Discuss briefly "combustion charts".
19. What are combustion charts ? Where these are used and why ?
20. Write a short note on gas tables.
21. Discuss the effect of the following variables on pressure and temperature at salient points of Otto cycle on the basis of fuel-air cycle.
    (i) Compression ratio (ii) Fuel-air ratio.
22. What is the difference between air cycle and fuel-air cycle ? What are the assumptions in fuel-air cycle ?
23. What is the use of fuel-air cycle ?
24. What is the difference between air standard cycles and fuel-air cycles.
25. Make a comparative statement of operations and working media for air cycle, fuel-air cycle and actual cycle of S.I. engines.
26. Explain why a S.I. engine fails to operate if the air-fuel ratio is more than 20 : 1 while a C.I. engine can operate on an air-fuel ratio of even 50 : 1.
27. Explain how (i) time losses and (ii) incomplete combustion losses are accounted for in the real-cycle analysis.
28. "Air-fuel ratio in a S.I. engine varies fr om 8 to 16 approximately while such variation in a C.I. engine is from 100 at no-load to 20 at full load". Explain.

UNSOLVED EXAMPLES

1. Find the change in efficiency of an Otto cycle for a compression ratio of 7, if the specific heat at constant volume increases by 1 percent. [Ans. – 0.663%]
2. The following data relate to a petrol engine :
   Compression ratio = 6
   Calorific value of fuel used = 44000 kJ/kg
   The air-fuel ratio = 15 : 1
   The temperature and pressure of the charge at the end of the stroke = 60°C, 1 bar
   Index of compression = 1.32
   The specific heat at constant volume, $c_v = 0.71 + 20 \times 10^{-5}\, T$ kJ/kg K where T is in K.
   Determine the maximum pressure in the cylinder. Compare this value with that of constant specific heat $c_v = 0.71$ kJ/kg K. [Ans. 56.6 bar ; 80.5 bar]
3. The combustion in a diesel engine is assumed to begin at inner dead centre and to be at constant pressure. The air-fuel ratio is 28 : 1, the calorific value of the fuel is 42 MJ/kg, and the specific heat of the products of combustion is given by :
   $c_v = 0.71 + 20 \times 10^{-5}\, T$; R for the products = 0.287 kJ/kg K.
   If the compression ratio is 14 : 1, and the temperature at the end of compression is 800 K, find at what percentage of the stroke combustion is completed. [Ans. 10.96% stroke]
4. In an oil engine working on dual combustion cycle the temperature and pressure at the beginning of compression are 90°C and 1 bar respectively. The compression ratio is 13 : 1. The heat supplied per kg of air is 1675 J, half of which is supplied at constant volume and half at constant pressure. Calculate :
   (i) The maximum pressure in the cycle ;
   (ii) The percentage of stroke at which cut-off occurs.
   Take : γ for compression = 1.4 ; R = 0.287 kJ/kg K and c_v for products of combustion = $0.71 + 20 \times 10^{-5}\, T$.
   [Ans. 66.2 bar ; 2.64% of stroke]

LNQM assumes *compression curve* having *no ignition.*

- First stage of combustion, the *ignition lag*, starts from this point and *no pressure rise is noticeable.*
- *Q* is the point where the *pressure rise can be detected.* From this point it deviates from the simple compression (motoring) curve.
- The time lag between first igniting of fuel and the commencement of the main phase of combustion is called the *period of incubation* or is also known as **ignition lag.** The time is normally about 0.0015 seconds. The maximum pressure is reached at about 12° after top dead centre point. Although the point of maximum pressure marks the completion of flame travel, it does not mean that at this point the whole of the heat of fuel has been liberated, for even after the passage of the flame, some further chemical adjustments due to reassociation, etc., will continue to a greater or less degree throughout the expansion stroke. This is known as ***after burning.***

Effect of engine variables on flame propagation :

1. ***Fuel-air ratio.*** When the *mixture is made leaner or is enriched* and *still more, the velocity of flame diminishes.*

2. ***Compression ratio.*** The *speed of combustion increases with increase of compression ratio.* The increase in compression ratio results in increase in temperature which increases the tendency of the engine to detonate.

3. ***Intake temperature and pressure.*** *Increase in intake temperature and pressure increases the flame speed.*

4. ***Engine load.*** As the *load on the engine increases,* the cycle pressures increase and hence the flame *speed increases.*

5. ***Turbulence.*** The flame speed is very low in non-turbulent mixture. A turbulent motion of the mixture intensifies the processes of heat transfer and mixing of the burned and unburned portions in the flame front. These two factors cause the *velocity of turbulent flame to increase practically in proportion to the turbulent velocity.*

6. ***Engine speed.*** The *flame speed increases almost linearly with engine speed.* The *crank angle required for flame propagation, which is the main phase of combustion, will remain almost constant at all speeds.*

7. ***Engine size.*** The number of crank degrees required for flame travel will be about the same irrespective of engine size, provided the engines are similar.

5.2.1.1. Factors affecting normal combustions in S.I. engines.

The factors which affect normal combustion in S.I. engines are briefly discussed below :

1. ***Induction pressure.*** As the pressure falls delay period increases and the ignition must be earlier at low pressures. A *vacuum control* may be incorporated.
2. ***Engine speed.*** As speed increases the constant time delay period needs more crank angle and ignition must be earlier. A *centrifugal control* may be employed.
3. ***Ignition timing.*** If ignition is too early the peak pressure will occur too early and work transfer falls. If ignition is too late the peak pressure will be low and work transfer falls. Combustion may not be complete by the time the exhaust valve opens and the valve may burn.
4. ***Mixture strength.*** Although the stoichiometric ratio should give the best results, the effect of dissociation shown in Fig. 5.3 is to make a slightly rich mixture necessary for maximum work transfer.
5. ***Compression ratio.*** An increase in compression ratio increases the maximum pressure and the work transfer.

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- The early ignition created by pre-ignition extends the total time and the burnt gases remain in the cylinder and therefore *increases the heat transfer on the chamber walls, as a result, the self-ignition temperature will occur earlier and earlier on each successive compression stroke.* Consequently, the **peak cylinder pressure** (which normally occurs at its optimum position of 10°–15° after T.D.C.) will progressively *advance* its position to T.D C. where the cylinder pressure and temperature will be maximised
  — The accumulated effects of an extended **combustion** time and rising **peak cylinder pressure and temperature** cause the **self-ignition temperature** to creep further and further ahead of T.D.C., and with it, **peak** cylinder pressure, which will now take place before T.D.C. so that **negative work** *will be done in compressing the combustion products* (Fig. 5.5).

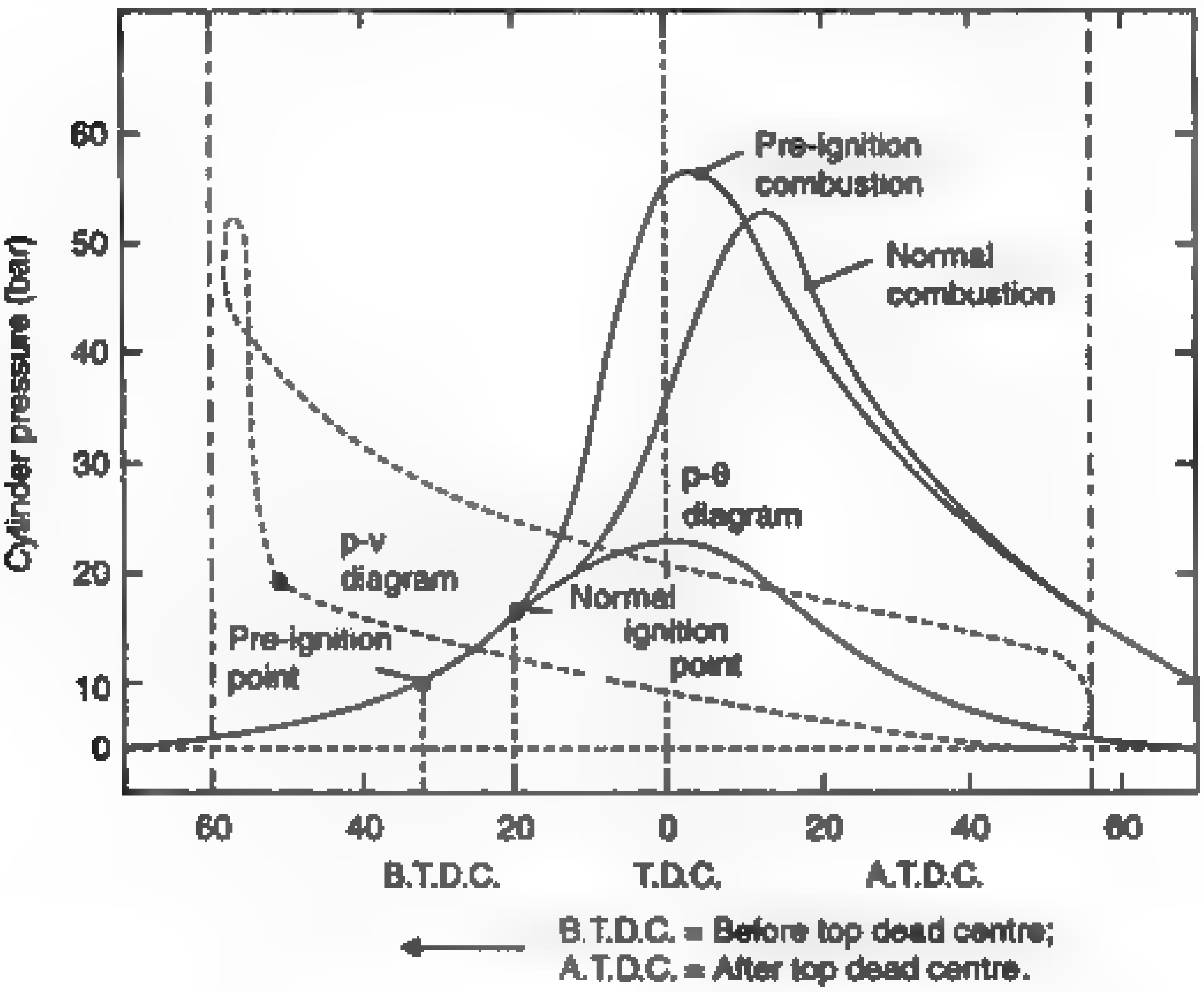

Fig. 5.5. Cylinder pressure variation when pre-ignition occurs.

Effects of pre-ignition :

1. It increases the tendency of detonation in the engines.

2. Pre-ignition is a serious type of abnormal combustion. It increases the heat transfer to the cylinder walls because high temperature gases remain in contact with the cylinder for a longer period. The load on the crankshaft during compression is abnormally high. This may *cause crank failure.*

3. Pre-ignition in a single-cylinder engine will result in a *steady reduction in speed and power output.*

4. The *real undesirable effects of pre-ignition* are *when it occurs only in one or more cylinders in a multi-cylinder engine.* Under these conditions, when the engine is driven hard, the unaffected cylinders will continue to develop their full power and speed, and so will drag the other piston or pistons, which are experiencing pre-ignition and are producing negative work, to and fro until *eventually the increased heat generated makes the pre-igniting cylinders' pistons and rings sieze.*

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4. **Ignition timing.** *Advanced ignition timing* increases peak pressures and *promotes* knock.

5. **Mixture strength.** *Optimum mixture strength* gives high pressures and *promotes* knock.

6. **Compression ratio.** *High compression ratios* increase the cylinder pressures and *promotes* knock.

7. **Combustion chamber design.** Poor design gives long flame paths, poor turbulence and insufficient cooling all of which *promote* knock.

8. **Cylinder cooling.** *Poor cooling* raises the mixture temperature and *promotes* knock.

5.7. PERFORMANCE NUMBER (PN)

Performance number is a *useful measure of detonation tendency*. It has been developed from the conception of knock limited indicated mean effective pressure (*klimep*), when inlet pressure is used as the dependent variable.

$$\text{Performance number (PN)} = \frac{klimep \text{ of test fuel}}{klimep \text{ of iso-octane}}$$

The performance number is *obtained on specified engine, under specified set of conditions by varying the inlet pressure.*

5.8. HIGHEST USEFUL COMPRESSION RATIO (HUCR)

The **highest useful compression ratio** *is the highest compression ratio employed at which a fuel can be used in a specified engine under specified set of operating conditions, at which detonation first becomes audible with both the ignition and mixture strength adjusted to give the highest efficiency.*

5.9. COMBUSTION CHAMBER DESIGN—S.I. ENGINES

Engine torque, power output and fuel consumption are profoundly influenced by the following :

(*i*) Engine compression ratio ;

(*ii*) Combustion chamber and piston crown shape ;

(*iii*) The number and size of the inlet and exhaust valves ;

(*iv*) The position of the sparking plug.

The following are the objects of good combustion chamber design :

1. *To optimize the filling and emptying of the cylinder with fresh unburnt charge respectively over the engine's operating speed range ; and*

2. *To create the condition in the cylinder for the air and fuel to be thoroughly mixed and then excited into a highly turbulent state so that the burning of the charge will be completed in the shortest possible time.*

3. To prevent the possibility of detonation at all times, as far as possible in order to achieve these fundamental requirements it is imperative to be aware of the *factors that contribute towards inducing the charge to enter the cylinder, to mix intimately, to burn both rapidly and smoothly and to expel the burnt gases.*

2. The increase of flame speed due to turbulence reduces the combustion time and hence minimises the tendency to detonate.
3. Turbulence increase the heat flow to the cylinder wall and in the limit excessive turbulence may extinguish the flame.
4. Excessive turbulence results in the more rapid pressure rise (though maximum pressure may be lowered) and the high pressure rise causes the crankshaft to spring and rest of the engine to vibrate with high periodicity, resulting in rough and noisy running of the engine.

5.9.5. Flame Propagation

- Typical flame propagation velocities range from something like 15 to 70 m/s. This would relate to the combuston flame velocity increasing from about 15 m/s at an idle speed of about 1000 r.p.m. to roughly 70 m/s at a maximum speed of 6000 r.p.m.
- *When ignition occurs the nucleus of the flame spreads with the whirling or rotating vortices in the form of ragged burning crust from the initial spark plug ignition site.*
- ***The speed of the flame propagation is roughly proportional to the velocity at the periphery of the vortices.***

5.9.6. Swirl Ratio

- Induction swirl can be generated by tangentially directing the air movement into the cylinder either by creating a preswirl in the induction port or by combining the tangential-directed flows with a preswirl helical port. *"Cylinder air swirl"* is defined *as the angular rotational speed about the cylinder axis.*
- ***Swirl ratio*** is defined as the *ratio of air rotational speed to crankshaft rotational speed.*
  — Helical ports can achieve swirl ratio of 3 to 5 at T.D.C. with a flat piston crown. However, if a bowl in the piston chamber is used, the swirl ratio can be increased to about 15 at T.D.C.

5.9.7. Surface-to-Volume Ratio

- In order to *minimise* the heat losses and formation of hydrocarbons within the combustion

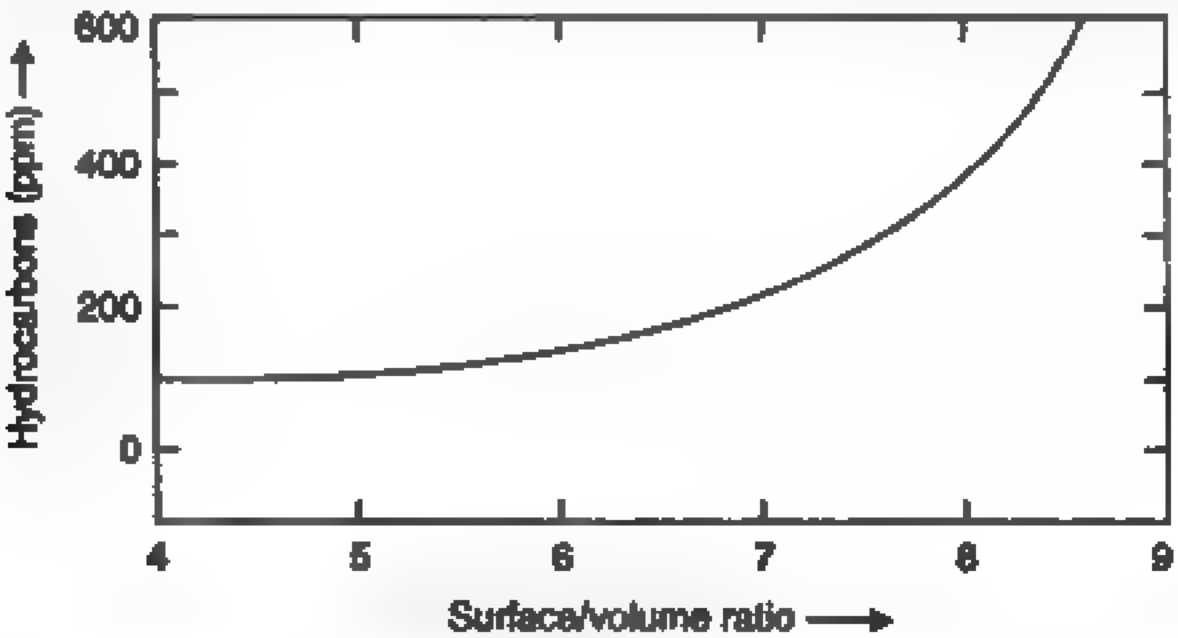

Fig. 5.13

chamber, the chamber volume should be maximised relative to its surface area, that is, the *chamber's surface area should be as small as possible relative to the volume occupied by the combustion chamber* (Fig. 5.13). The surface-to-volume ratio is the *ratio of the combustion surface area to that of its volume.*

- The surface-to-volume ratio *increases linearly with rising compression ratio.*

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— Also, because of heterogeneous mixture, lean mixture is used

These factors make the engine heavier

— The *incomplete combustion of heterogeneous mixture, and droplet combustion result in the smoke and odour.*

- C.I. engines are manufactured in the following range of speeds, speeds and power outputs :

Particulars	*Range*
1. Piston diameters	50 mm to 900 mm
2. Speeds	100 r.p.m. to 4400 r.p.m
3. Power output	2 B.P. to 40000 B.P.

6.2. COMBUSTION PHENOMENON IN C.I. ENGINES

- The process of combustion in the compression ignition (C.I.) engine is fundamentally different from that in a spark-ignition engine. In C.I. engine combustion occurs by the high temperature produced by the compression of the air, *i.e.* it is an *auto-ignition.* For this a minimum compression ratio of 12 is required. The efficiency of the cycle increases with higher values of compression ratio but the maximum pressure reached in the cylinder also increases. This requires heavier construction. The upper limit of compression ratio in a C.I. engine is due to mechanical factor and is a compromise between high efficiency and low weight and cost. The normal compression ratios are in the range of 14 to 17, but may be upto 23. The air-fuel ratios used in the C.I. engine lie between 18 and 25 as against about 14 in the S.I. engine, and *hence C.I. engines are bigger and heavier for the same power than S.I. engines.*
- In the C.I. engine, the intake is air alone and the fuel is injected at high pressure in the form of fine droplets near the end of compression. This leads to delay period in the C.I. engine, is greater than that in the S.I. engine. The *exact phenomenon of combustion in the C.I. engine* is explained below.

  — Each minute droplet of fuel as it enters the highly heated air of engine cylinder is quickly surrounded by an envelope of its own vapour and this, in turn and at an appreciable interval is inflamed at the surface of the envelope. To evaporate the liquid, latent heat is abstracted from the surrounding air which reduces the temperature of the thin layer of air surrounding the droplet, and some time must elapse before this temperature can be raised again by abstracting heat from the main bulk of air in this vicinity. As soon as this vapour and the air in actual contact with it reach a certain temperature, ignition will take place. Once ignition has been started and a flame established the heat required for further evaporation will be supplied from that released by combustion The vapour would be burning as fast as it can find fresh oxygen, *i.e., it will depend upon the rate at which it is moving through the air or the air is moving past it.*

  — In the C.I. engine, the fuel is not fed in at once but is spread over a definite period. The first arrivals meet air whose temperature is only a little above their self-ignition temperature and the delay is more or less prolonged. The later arrivals find air already heated to a far higher temperature by the burning of their predecessors and therefore light up much more quickly, almost as they issue from the injector nozzle, but their subsequent progress is handicapped for there is less oxygen to find.

  — If the air within the cylinder were motionless, only a small proportion of the fuel would find sufficient oxygen, for it is impossible to distribute the droplets uniformly throughout the combustion space. Therefore some air movement is absolutely essential, as in the S.I. engine. But there is a fundamental difference between the

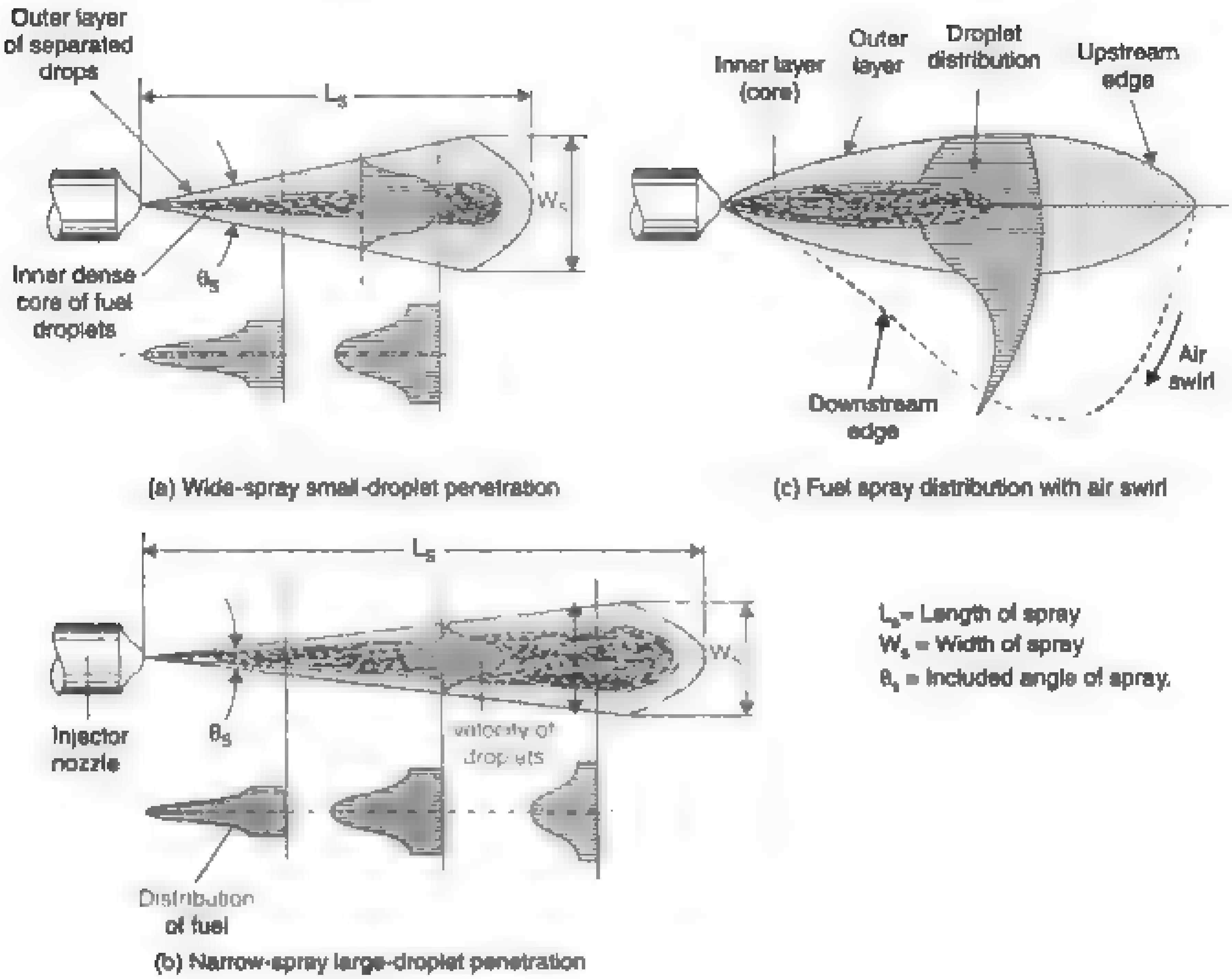

Fig 6.2. Injected fuel spray characteristics.

- The liquid core, now surrounded by layers of heated vapour, oxidises burns as fast as it can ; that is it finds fresh oxygen to keep the chemical reaction going on.
- When the ***physical delay*** to convert the fuel spray into tiny droplets and the ***chemical reaction delay*** to establish ignition from the initial oxidation process are over, *the rate of burning is dependent on the speed at which the droplets are moving through the air or the air is moving past the droplets.*

Compression Ratio (r) :

Increase in compression ratio exercises the following *effects* :

- The cylinder compression pressure and temperature *increase* ; the ignition time lag between the point of injection to the instant when ignition first commences *reduces*.
- The density and turbulence of the charge *increase*, and this *increases the rate of burning and, accordingly the rate of pressure rise and the magnitude of the peak cylinder pressure reached.* The characteristics of the pressure rise relative to the piston stroke or crank-angle movement is illustrated in Fig 6.3 and Fig 6.4.
- Thermal efficiency and the specific fuel consumption are *improved* (Fig 6.5)
- Raising compression ratio *results in reduction in the mechanical efficiency* as shown in Fig. 6.6 (since the higher cylinder pressures increase the pumping losses, friction

Effects of Various Factors on Delay Period :

Effects of various factors such as *fuel properties, intake temperature,* compression ratio, engine speed, type of combustion chamber, and injection advance are discussed below :

1. **Fuel properties :**

- The *self-ignition temperature (S.I.T.)* is the most important property of the fuel which affects the delay period.
  - — *A lower S.I.T. means a wide margin between it and the temperature of compressed air and hence lower delay period.*
  - — *Higher cetane number means a lower delay period and smoother engine operation.* Cetane number depends on the *chemical composition of fuel.* The *more paraffinic hydrocarbons are contained in fuel, higher will be the cetane number*
- The other fuel properties which affect delay period are :
  (*i*) Volatility ;
  (*ii*) Latent heat ;
  (*iii*) Viscosity ;
  (*iv*) Surface tension.
  - — *Volatility and latent* heat affect the *time taken to form an envelope of vapour.*
  - — The *viscosity and surface tension* influence the *fineness of atomisation.*

2. **Intake temperature :**

- *Increase in intake temperature* would result in increase in compressed air temperature which would *reduce the delay period.*

3. **Compression ratio :**

- *Increase in compression ratio reduces delay period as it raises both temperature and density*
  - — With increase in compression ratio, temperature of air increases. At the same time the minimum auto-ignition temperature decreases due to increased density of compressed air resulting in closer contact of molecules which thereby reduces the time of reaction when fuel is injected.
  - — As *the difference between compressed air temperature and minimum auto-ignition temperature increases, the delay period decreases.*

4. **Engine speed :**

- Delay period can be given either in terms of *absolute time* (in milliseconds) *or crank angle rotation.*
  - — At *constant speed*, delay period *is proportional to the delay angle.*
  - — In *variable speed operation*, delay period may *decrease in terms of milliseconds but increase in terms of crank angles.*

5. **Type of combustion chamber :**

- *A pre-combustion chamber gives shorter delay compared to an open type of combustion chamber.*

6. **Injection advance :**

- *Delay period increases with increase in injection advance angle.* The reason for increase in delay period with increase in injection advance angle is that pressures and temperatures are lower when injection begins.

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Advantages of induction swirl :

1. Easier starting (due to low intensity of swirl).
2. High excess air (low temperature), low turbulence (less heat loss), therefore indicated thermal efficiency is high.
3. Production of swirl requires no additional work.
4. Used with low speeds, therefore low quality of fuel can be used.

Disadvantages :

1. Shrouded valves, smaller valves, low volumetric efficiency.
2. Weak swirl, low air utilisation (60%), lower m.e.p. and large size (costly) engine.
3. Weak swirl, multi-orifice nozzle, high induction pressure, clogging of holes, high maintenance.
4. Swirl not proportional to speed ; efficiency not maintained at variable speed engine.
5. Influence minimum quantity of fuel. Complication at high loads and idling.

Compression swirl :

- The second method of generating swirl is by compression swirl in what is known as *swirl chamber*. A swirl chamber is a *divided chamber*. A divided combustion chamber is defined as one in which combustion space is divided into two or more distinct compartments, between which there are restrictions or throats small enough so that considerable pressure differences occur between them during combustion process.

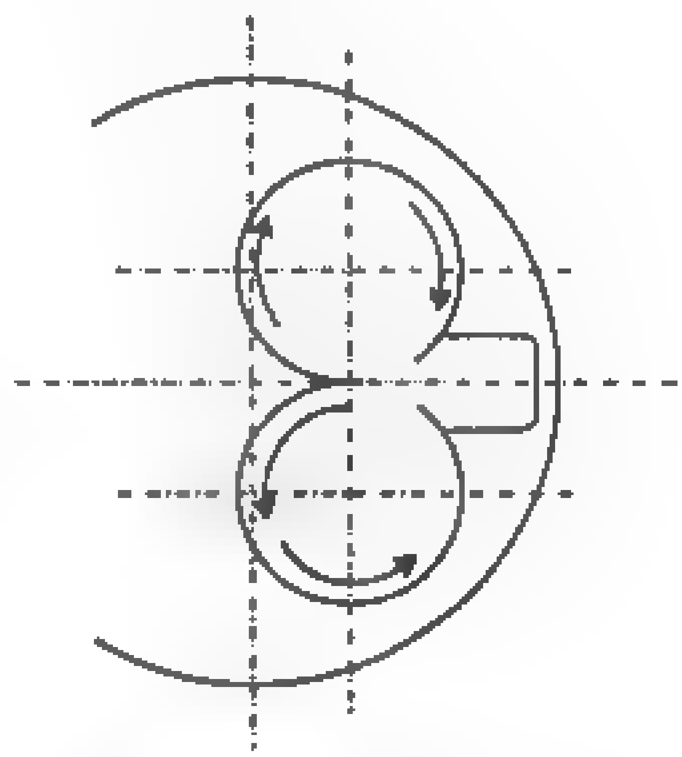

Fig. 6.9. Compression swirl.

- This swirl is maximum at about 15° before T.D.C. *i.e.* close to the time of injection. The fuel is injected into the swirl chamber and ignition and bulk of combustion takes place therein. A considerable amount of heat is lost when products of combustion pass back through the some throat and this loss of heat is reduced by employing a heat insulated chamber. Thus, it serves as a thermal regenerator receiving heat during combustion and expansion and returning the heat to air during compression stroke. However the loss of heat to surface of combustion chamber is greater than induction swirl.
- In *combustion swirl, a very strong swirl which increases with speed is generated.*

Advantage of compression swirl :

1. Large valves, high volumetric efficiency.
2. Single injector, pintle type (self cleaning), less maintenance.
3. Smooth engine operation.
4. Greater air utilization due to strong swirl. Smaller (cheaper) engine.
5. Swirl proportional to speed, suitable for variable speed operation.

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progresses at a more rapid rate. The pressures built up in the minor cell, therefore, force the burning gases out into the main combustion chamber, thereby creating added turbulence and producing better combustion in this chamber. In the mean time, pressure is built up in the major cell, which then prolongs the action of the jet stream entering the main chamber, thus continuing to induce turbulence in the main chamber.

5. M. Combustion chamber :

- After twenty years of research in 1954, Dr. Meuner of M.A.N., Germany developed M-process engine which ran without typical diesel combustion noise and hence it was named *'whisper engine'*.
- Fig. 6.15 shows a combustion chamber developed for small high speed engines. It differs from the other open combustion chamber engines in the respect that *fuel spray impinges tangentially on, and spreads over, the surface of a spherical space in the piston.* There is always some impingement of spray on the combustion chamber walls in all successful diesel engine designs. This impingement was not considered desirable till M.A.N. combustion system was experimented.
- The M.A.N. system's theory is that *enough of spray will ignite before impingement so that delay period will be normal while most of the fuel spray will evaporate from the hemispherical combustion space in piston prior to combustion.* Thus the *second stage of combustion is slowed down avoiding excessive rate of pressure rise.* Shrouded inlet valve is used to give air swirl in direction of arrow.

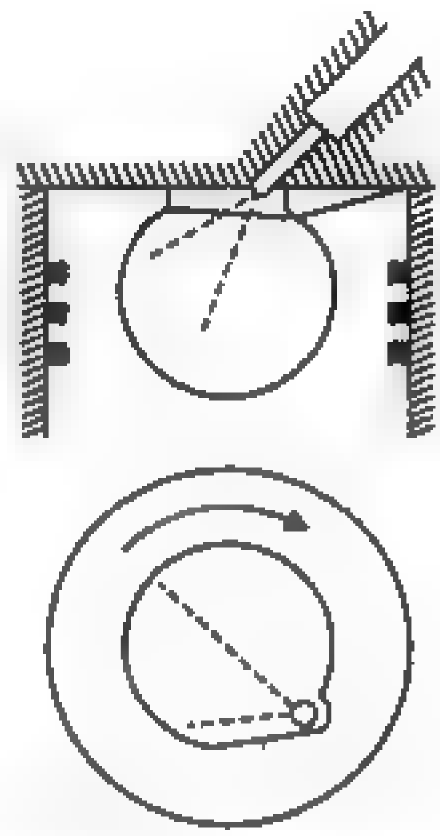

Fig. 6.15. M.A.N. 'M' combustion chamber.

Advantages :

'M-chamber' claims the following *advantages :*

(*i*) Low peak pressure.

(*ii*) Low rate of pressure rise.

(*iii*) Low smoke level.

(*iv*) Ability to operate on a wide range of liquid fuels (multi-fuel capability).

Disadvantages :

(*i*) Low volumetric efficiency.

(*ii*) Since fuel vaporisation depends upon the surface temperature of the combustion chamber, cold starting requires certain aids.

(*iii*) At starting and idling conditions hydrocarbon emissions may occur.

Table 6.1 gives *comparison between open combustion chambers and divided combustion chambers.*

S. No.	*Aspects*	*Open Combustion Chamber*	*Divided Combustion Chamber*
1.	*Fuel used*	Can consume fuels of good ignition quality, i.e. of shorter ignition delay or higher cetane number	Can consume fuels of poor ignition quality *i.e.*, larger ignition delay. or lower cetane number.
2.	*Type of injection nozzle used*	Requires multiple hole injection nozzles for proper mixing of fuel	It is able to use single hole injection nozzles and moderate injection

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12. Explain briefly the phenomenon of "Diesel knock".
13. State the differences in the knocking phenomena of S.I. and C.I. engines.
14. Enlist various methods of controlling diesel knock.
15. What should be the primary considerations in the design of combustion chambers for C.I. engines ?
16. Explain briefly basic methods of generating air swirl in C.I. engines combustion chambers.
17. Enlist the advantages and disadvantages of induction swirl.
18. State the advantages and disadvantages of compression swirl.
19. Explain briefly any two of the following combustion chambers :
    (*i*) Open or direct combustion chamber (*ii*) Turbulent chamber
    (*iii*) Pre-combustion chamber (*iv*) Energy cell.
20. Give the comparison between open combustion chambers and divided combustion chambers.
21. Write short note on cold starting of C.I. engines.
22. Explain briefly cold starting aids for C.I. engines.
23. Explain the phenomenon of knock in C.I. engines and compare it with S.I. engine knock.
24. How does the mixture composition in combustion chamber of a C.I. engine differ from that of a S.I. engine ?
25. "The factors that tend to increase detonation in S.I. engine tend to reduce knocking in C.I. engine". Discuss the above statement with reference to the following influencing factors :
    (*i*) Compression ratio ; (*ii*) Inlet temperature ;
    (*iii*) Inlet pressure ; (*iv*) Self-ignition temperature of fuel ;
    (*v*) Time lag of ignition of fuel ; (*vi*) r.p.m. ;
    (*vii*) Combustion chamber wall temperature.
26. Why does rate of pressure rise during combustion is limited to a certain value ?
27. Discuss the influence of ignition delay on combustion processes in S.I. and C.I. engines. Explain how the presence of a knock inhibitor in fuel oil helps to change the ignition delay in C.I. engines.
28. "The requirement of air motion and swirl in a C.I. engine combustion chamber is more strigent than in a S.I. engine". Justify the statement.
29. "The induction swril in a C.I. engine helps in increasing indicated thermal efficiency". Justify the statement.
30. How are C.I. engine combustion classified ? What type of swirl is used in these chambers ?
31. "In agriculture field, it is better to use C.I. engine than S.I. engine". Justify the statement.
32. How can a diesel engine be converted to CNG engine ?
33. "The maximum substitution of diesel engine by CNG in a C.I. engine is limited by the cetane characteristics of the available fuel". Justify the statement.
34. Write a short note on aids for starting C.I. engines under extreme cold climate.
35. Describe the M-combustion system and discuss its relative merits with respect to D.I. chambers.

- *Multipoint injectors* which add fuel at the *intake valve ports* will have *better efficiency* because the air is displaced until after the intake manifold. Fuel evaporation does not occur until the flow is entering the cylinder at the intake valve
- *Those engines that inject fuel directly into the cylinders after the intake valve is closed will experience no volumetric efficiency loss due to fuel evaporation.* Manifolds with late fuel addition may be designed to further increase volumetric efficiency by having large diameter runners. High velocity and turbulence to promote evaporation are *not needed*. They can also be operated cooler, which results in a *dense inlet air flow*.
- Fuels like *alcohol* which have a *smaller air-fuel ratio* will experience a *greater loss in volumetric efficiency*. Fuels with high heat of vaporisation will regain some of this lost efficiency due to the greater evaporation cooling that will occur with these fuels. This cooling will create a denser air-fuel flow for a given pressure, allowing for more air to enter the system. Alcohol has high heat of vaporisation, so some efficiency lost due to *air-fuel* is gained back again.
- Gaseous fuels like hydrogen and methane *displace more incoming* air than liquid fuels, which are only partially evaporated at the intake system. This must be considered when trying to modify engines made for gasoline fuel to operate on these gaseous fuels. It can be assumed that fuel vapour pressure in the intake system is between 1 to 10 percent of total pressure when gasoline-type liquid fuel is being used. When gaseous fuels or alcohol is being used, the fuel vapour pressure is often *greater* than 10 percent of the total. *Intake manifolds can be operated much cooler when gaseous fuel is used, as no vapourisation is required.* This will gain back some lost volumetric efficiency.
- *The later that fuel vaporises in the intake system, the better is the volumetric efficiency.* On the other hand, *the earlier that fuel vaporises, the better are the mixing process and cylinder-to-cylinder distribution consistency.*

2. Heat transfer-High temperature :

- All intake systems are hotter than the surrounding air temperature and will consequently heat the incoming air. This *lowers the density of the air, which reduces volumetric efficiency.*
- Intake manifolds of carburetted systems or throttle body injection systems are *purposely heated to enhance fuel evaporation.* At lower engine speeds, the air flow rate is slower and the air remains in the intake system for a longer time. It thus gets heated to higher temperatures at low speeds, which *lowers the volumetric efficiency curve* in Fig. 7.2 at the low-speed end.
- Some systems have been tried *which inject small amounts of water into the intake manifold.* This is *to improve the volumetric efficiency by increasing the resulting evaporative cooling that occurs.*

3. Valve overlap :

- At the top dead centre (T.D.C.) at the end of exhaust stroke and the beginning of the intake stroke, both intake and exhaust valves are open simultaneously for a brief moment. When this happens, *some exhaust gas can get pushed through the open intake valve back into the intake system.* The *exhaust then gets carried back into the cylinder with the intake air-fuel charge, displacing some of the incoming air and lowering volumetric efficiency.* This *problem is greatest at low engine speeds, when real time of valve overlap is greater.* This effects *lowers efficiency curve* in Fig. 7.2 at the low engine speed end.
- Other factors that affect the above problem are the *intake and exhaust valve location and compression ratio.*

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Solution. *Given :* $D = 20.3$ cm $= 0.203$ m ; $L = 30.5$ cm $= 0.305$ m ; $N = 300$ r.p.m.

$\eta_{vol.} = 78\%$, A/F ratio $= 4 : 1$

Volume of gas used per minute :

$$\text{Stroke volume} = \frac{\pi}{4} \times 0.203^2 \times 0.305 = 0.009871 \text{ m}^3$$

$$\text{Volume inhaled} = \eta_{vol.} \times \text{stroke volume}$$
$$= 0.78 \times 0.009871 = 0.007699 \text{ m}^3$$

$$\text{Gas inhaled} = \frac{0.007699}{4+1} = 0.00154 \text{ m}^3$$

$$\text{Gas inhaled per minute} = 0.00154 \times \frac{300}{2} = \mathbf{0.231\ m^3/min.} \quad \textbf{(Ans.)}$$

Example 7.2. *A four-stroke, eight-cylinder engine is tested while running at 3600 r.p.m. The inlet air temperature is 15°C and the pressure is 760 mm of Hg. The total piston displacement volume is 4066 cm³. The air-fuel ratio of the engine is 14 : 1 and b.s.f.c. is 0.38 kg/kWh. Dynometer reading shows a power output of 86 kW. Find the volumetric efficiency of the engine.*

Solution. *Given :* $N = 3600$ r.p.m. ; Inlet temp. $T = 15°C$ or 288 K ;

$p = 760$ mm Hg $= 1.013$ bar

$V_s = 4066$ cm³ or 4066×10^{-6} m³ ; A/F ratio $= 14 : 1$;

b.s.f.c. $= 0.38$ kg/kWh

B.P. $= 86$ kW

Volumetric efficiency, $\eta_{vol.}$:

$$\text{Air consumption,} \quad m = \frac{86 \times 0.38 \times 14}{60} = 7.625 \text{ kg/min}$$

Also, $pV = mRT$

or
$$V = \frac{mRT}{p} = \frac{7.625 \times 287 \times 288}{1.013 \times 10^5} = 6.222 \text{ m}^3/\text{min}$$

$$\text{Displacement or swept volume} = 4066 \times 10^{-6} \times \frac{3600}{2}$$
$$= 7.319 \text{ m}^3/\text{min}$$

$$\therefore \quad \eta_{vol.} = \frac{6.222}{7.319} = 0.85 \text{ or } \mathbf{85\%.} \quad \textbf{(Ans.)}$$

Example 7.3. *The airflow to a four-cylinder, four-stroke oil engine is measured by a 5 cm diameter orifice having a coefficient of discharge of 0.6. The engine having bore 10 cm and stroke 12 cm runs at 1200 r.p.m. Pressure drop across orifice is 4.6 cm of water and ambient temperature and pressure are 17°C and 1 bar respectively. Calculate the volumetric efficiency based on free air condition.*

Solution. *Given :* $n = 4$; $d = 5$ cm $= 0.05$ m; $C_d = 0.6$, $D = 10$ cm $= 0.1$ m ; $L = 12$ cm $= 0.12$ m ; $N = 1200$ r.p.m. ; $h_w = 4.6$ cm $= 0.046$ m ; $T = 17 + 273 = 290$ K ; $p = 1$ bar.

$\eta_{vol.}$:

$$\rho_a = \frac{p}{RT} = \frac{1 \times 10^5}{287 \times 290} = 1.2015 \text{ kg/m}^3$$

Head causing flow, metre of air,

$$h_a = \frac{h_w \rho_w}{\rho_a} = \frac{0.046 \times 1000}{1.2015} = 38.285 \text{ m}$$

For change in inlet temperature :

$$\frac{\eta_{v_2}}{\eta_{v_1}} = \sqrt{\frac{T_{i_2}}{T_{i_1}}} = \sqrt{\frac{333}{300}} = 1.054$$

$\therefore$ **For both change in inlet pressure and temperature,**

$$\frac{\eta_{v_2}}{\eta_{v_1}} = 1.0125 \times 1.054 = 1.067$$

The percentage increase in volumetric efficiency = **6.7%. (Ans.)**

(*ii*) **The percentage change in indicated output of the engine :**

$$\text{Output} = \rho \times \eta_v$$

$$\rho_1 = \frac{p_1}{RT_1} = \frac{1.013 \times 10^5}{287 \times 300} = 1.176$$

$$\rho_2 = \frac{p_2}{RT_2} = \frac{1.3 \times 10^5}{287 \times 333} = 1.36 \text{ kg/m}^3$$

$$\therefore \quad \frac{P_2}{P_1} = \frac{\rho_2 \times \eta_{v_2}}{\rho_1 \times \eta_{v_1}} = \frac{1.36}{1.176} \times 1.067 = 1.234$$

or *Percentage increase in power* = **23.4%. (Ans.)**

Example 7.8. *A petrol engine operating at full throttle develops 32 kW with 80 percent mechanical efficiency at sea level where atmospheric conditions are 1.013 bar pressure and 35°C temperature. The engine is moved to a hill station whose altitude is 2000 m and temperature is 5°C. A drop of 10 mm of mercury barometer reading may be assumed for each 100 m of rise in altitude.*

Determine the percentage change in volumetric efficiency of the engine and the brake power of the engine if it runs at the same speed and full throttle.

Solution. *Given :* $(\text{I.P.})_1 = 32$ kW ; $\eta_{mech.} = 80\%$; $p_1 = 1.013$ bar ; $T_1 = 35 + 273 = 308$ K ;
$T_2 = 5 + 273 = 278$ K

Percentage change in volumetric efficiency :

The ratio of p_e/p_i in both cases is 1. Therefore there is no change in volumetric efficiency due to inlet and exhaust pressure change.

For inlet temperature change,

$$\frac{\eta_{v_2}}{\eta_{v_1}} = \sqrt{\frac{T_{i_2}}{T_{i_1}}} = \sqrt{\frac{278}{308}} = 0.95$$

$\therefore$ *Percentage decrease in* $\eta_v = \dfrac{1 - 0.95}{1} \times 100 = 5\%$. **(Ans.)**

Brake power of the engine, $(\text{B.P.})_2$:

Indicated power output (I.P.) $= \eta_v \times \rho_i$ or $\dfrac{\eta_v \, p_i}{T_i}$

$$(\text{I.P.})_1 = \frac{\eta_{v_1} \, p_{i1}}{T_{i_1}} = \frac{\eta_{v_1} \times 1.013}{308} = \eta_{v_1} \times 0.003289$$

Drop in pressure at hill station $= \rho g h$ N/m²

$$= (13.6 \times 1000) \times 9.81 \times \left(\frac{10}{1000} \times \frac{2000}{100}\right) \times 10^{-5} \text{ bar} = 0.267 \text{ bar}$$

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Volume of 1 kg of petrol vapour $= \dfrac{mRT}{p}$

$$= \frac{1 \times 85.67 \times (273 + 17)}{0.98 \times 10^5} = 0.2535 \text{ m}^3$$

Volume of 18.835 kg of air $= \dfrac{mRT}{p}$

$$= \frac{18.835 \times 287 \times 290}{0.98 \times 10^5} = 15.996 \text{ m}^3$$

$\therefore$ Volume of mixture per kg of petrol

$$= 0.2535 + 15.996 = 16.25 \text{ m}^3. \quad \textbf{(Ans.)}$$

(*ii*) Now, *volumetric efficiency*

$$= \frac{\text{Volume of air per min at intake condition}}{\text{Swept volume per min}}$$

$$= \frac{16.25 \times (31/60)}{6 \times \frac{\pi}{4}(0.125)^2 \times 0.19 \times \frac{1600}{2}}$$

$$= 0.75 \quad \text{or} \quad \textbf{75\%}. \quad \textbf{(Ans.)}$$

Example 7.12. *On testing a spark ignition engine it was observed that the volumetric efficiency is maximum when inlet valve Mach Index is 0.55 and the indicated torque, and indicated mean effective pressure occured at maximum volumetric efficiency.*

The engine having a bore of 110 mm and stroke 140 mm produces maximum indicated torque when running at 2400 r.p.m.

(i) Determine the nominal diameter of the inlet valve.

(ii) If the same engine is required to develop maxmum indicated power at 2800 r.p.m., how will the inlet valve size be modified ?

(iii) If the same engine runs at 2800 r.p.m. without any inlet valve modifications, how will volumetric efficiency get affected ?

Pressure at intake valve = 0.88 bar ; Temperatue at intake valve = 340 K ; Inlet valve flow coefficient = 0.33.

Assume : Fuel-air mixture as perfect gas with $\gamma = 1.4$ and $R = 287$ J/kg K.

(iv) What would be volumetric efficiency at maximum power speed of 4800 r.p.m., for unmodified engine.

Solution. *Given* : $Z = 0.55$; $D_{cy} = 110$ mm $= 0.11$ m, $L = 140$ mm $= 0.14$ m ; $N = 2400$ r.p.m.; $p = 0.88$ bar, $T = 340$ K ; $K_i = 0.33$; $\gamma = 1.4$, $R = 287$ J/kg K.

(*i*) **Nominal diameter of inlet valve, D_{iv} :**

For the properties of mixture given in the data, the local sonic velocity of mixture of air-fuel at the inlet or suction valve is given by :

$$U_s = \sqrt{\gamma RT} = \sqrt{1.4 \times 287 \times 340} = 369.6 \text{ m/s}$$

Also,

$$\frac{U_g}{U_s} = \left(\frac{D_{cy}}{D_{iv}}\right)^2 \times \frac{U_p}{K_i U_s} = Z \qquad \text{...[Eqn. (7.9)]}$$

or

$$\left(\frac{D_{cy}}{D_{iv}}\right)^2 \times \frac{(2LN/60)}{K_i U_s} = Z \qquad \left[\because U_p = \text{piston speed} = 2LN/60\right]$$

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8

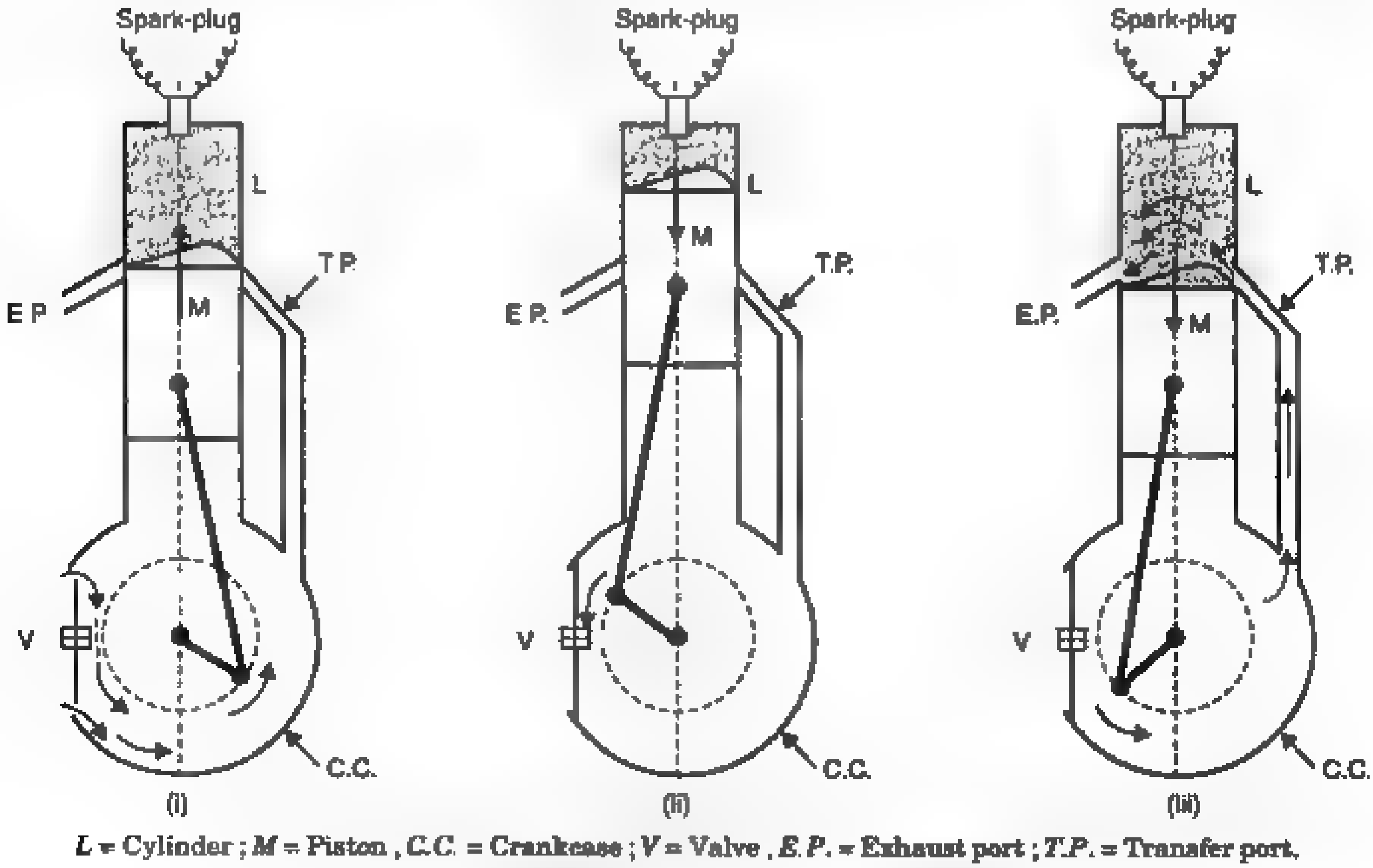
Two Stroke Engines

8.1. General aspects—Construction and working—Comparison between two stroke cycle and four stroke cycle engines—Disadvantages of two stroke S.I. engine compared to two stroke C.I. engine—Reasons for use of two stroke engines for marine propulsion—Reasons for the use of two stroke S.I. engines for low horse power two wheelers 8.2. Intake for two stroke cycle engines. 8.3. Scavenging process. 8.4. Scavenging parameters. 8.5. Scavenging systems. 8.6. Crankcase scavenging. 8.7. Scavenging pumps and blowers—Highlights—Objective Type Questions—Theoretical Questions.

8.1. GENERAL ASPECTS

8.1.1. Construction and Working

- In 1878, Dugald-clerk, a British engineer introduced a cycle which could be completed in two *strokes of piston rather than four strokes* as is the case with the four stroke cycle engines. The engines using this cycle were called *two stroke cycle engines*. In this engine suction and exhaust strokes are eliminated. Here *instead of valves, ports are used. The exhaust gases are driven out from engine cylinder by the fresh charge of fuel entering the cylinder nearly at the end of the working stroke.*
- Fig. 8.1 shows a two-stroke petrol engine (used in scooters, motor cycle etc.) Refer Art. 2.12 also.



L = Cylinder ; *M* = Piston , *C.C.* = Crankcase ; *V* = Valve , *E.P.* = Exhaust port ; *T.P.* = Transfer port.

Fig. 8.1. Two stroke cycle engine (crankcase scavenged).

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- There are open combustion chambers in the two stroke cycle engines. It would be extremely difficult to get proper scavenging in a cylinder with a divided chamber.
- In some automobile engines *standard-type superchargers are used and the air is input through intake valves with no fuel added.* The compressed air scavenges the cylinder and leaves it filled with air and a small amount of exhaust residual. After the intake valve is closed, fuel is injected directly into the combustion chamber by injectors mounted in the cylinder head. This is done to avoid HC pollution from fuel passing into the exhaust system, when both exhaust and intake valves are open. *In some automobile engines, air is injected with the fuel. This speeds evaporation and mixing, which is required because of the very short time of the compression stroke.*
  - Fuel injection pressure is of order of 500 to 600 kPa, while air injection pressure is slightly less at about 500 kPa.
  - *For "S.I. engine" fuel injection occurs early in the compression stroke, immediately after the exhaust valve closes.* In *"C.I. engines" the injection occurs late in the compression stroke, a short time before combustion starts.*
- In just about all two stroke cycle engines, *due to cost, crankcase compression is used to force air into and scavenge the cylinders.*
  - In these engines, air is introduced at atmospheric pressure into the cylinder below the piston through a one-way valve when the piston is near T.D.C. The power stroke pushes the piston down and compresses the air in the crankcase, which has been designed for this dual purpose. The compressed air then passes through an input channel into the combustion chambers. In *modern automobiles engines* the fuel is then added with injectors, as with supercharged engines the fuel is then added with injectors, as with supercharged engines. In *small engines the fuel is usually with a carburettor to the air as it enters the crankcase. This is done to keep the cost down on* small engines, simple carburettors being cheap to build. The fuel injectors will probably become more common as pollution laws become more stringent.
- In case of two stroke cycle engines *using crankcase compression, lubricating oil must be added to the inlet air.* The crankcase in these engines cannot be used as the oil reservoir as with most other engines. Instead, the surfaces of the engine components are lubricated by *oil vapour carried by the intake air.* In some engines, lubricating oil is mixed directly with the fuel and is vaporised in the carburettor along with the fuel. Other engines have a separate oil reservoir and feed lubricant directly into the intake air flow. *Two negative results* occur beacuse of this method of lubrications : (*i*) Some oil vapour gets into the exhaust flow during valve overlap and contributes directly to HC exhaust emissions ; (*ii*) Combustion is less efficient due to the poorer fuel quality of the oil.
  - *Engines which use superchargers or turbochargers generally use standard pressurised lubrication systems, with crankcase serving as the oil reservoir.*
- *In order to avoid an excess of exhaust residual no pockets of stagnant flow or dead zones can be allowed in the scavenging process. This is controlled by :*

  (*i*) The size and position of the intake and exhaust slots or valves ;

  (*ii*) The geometry of the slots in the wall ;

  (*iii*) The contoured flow deflectors on the piston face.

8.3. SCAVENGING PROCESS

- In a two stroke engine because of non-availability of an exhaust stroke (unlike four-stroke engine) at the end of an expression stroke, its combustion chamber is left full of

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Pressure loss co-efficient. It is defined as the *ratio between the main upstream and downstream pressures during the scavenging period and represents the pressure loss to which the scavenge air is subjected when it crosses the cylinder.*

Excess air factor (λ). The value (R_{del} –1) is called the excess air factor. Thus if the R_{del} (delivery ratio) is 1.3 the excess air factor is 0.3.

Measurement of Scavenging Efficiency.

The following procedure is adopted in *diesel engines for measuring the scavenging efficiency :*

- A small sample of the combustion products is drawn just before the exhaust valve opens or during the earlier part of blowdown.
- The sample is analysed.
- The results obtained are compared with standard curves of exhaust products *vs.* *F/A* ratio. This determines the *F/A* ratio that must have existed in the cylinder before combustion.
- Knowing the quantity of fuel injected per cycle, the quantity of fresh air retained in the cylinder per cycle is determined. Air present in the residual gas is not considered as it represent a constant quantity which does not participate in combustion process.

8.5. SCAVENGING SYSTEMS

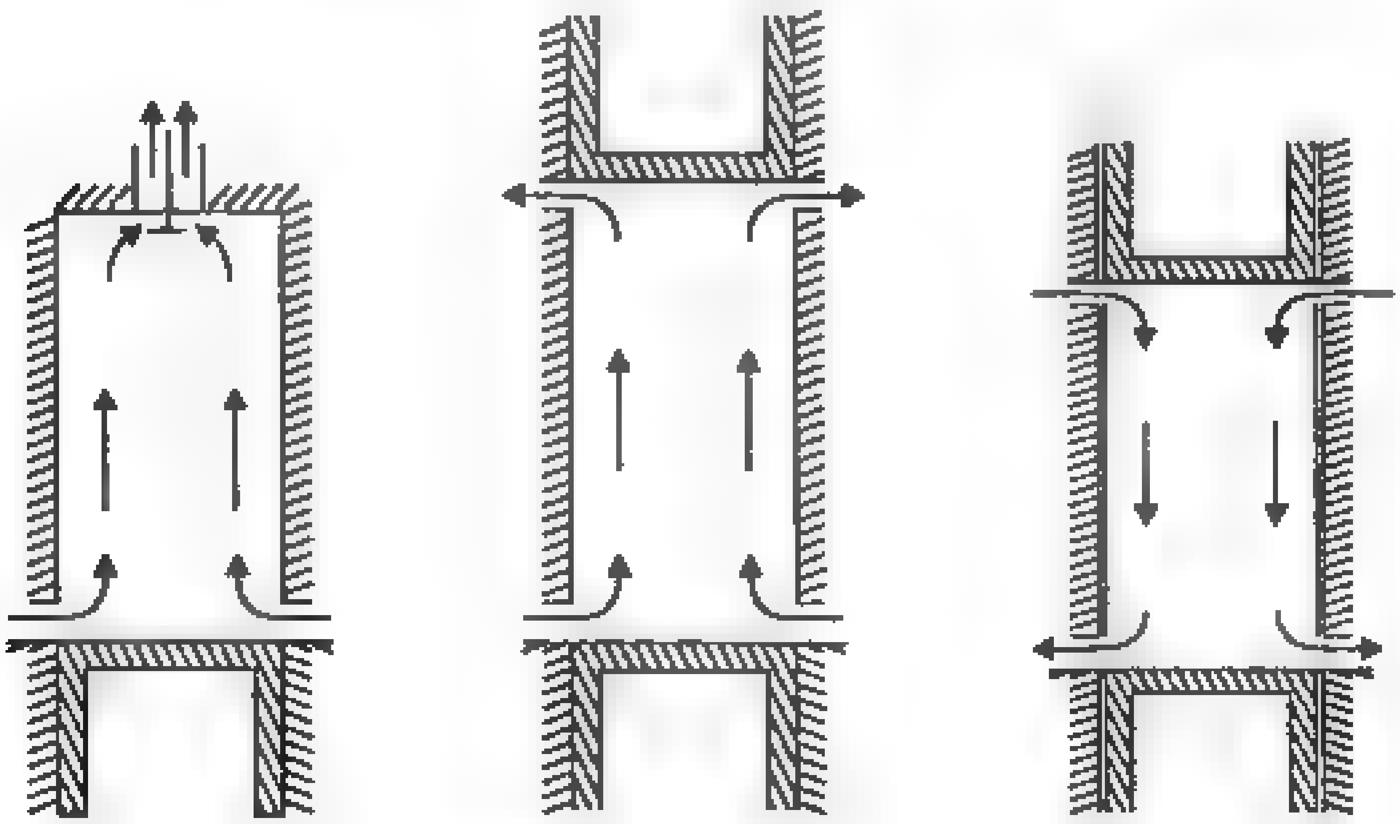
Different scavenging systems/arrangements *based on charge flow* are enumerated and described below :

1. Uniflow scavenging
2. Loop or reverse scavening
3. Cross scavenging.

1. Uniflow scavenging :

It is the *most perfect method of scavenging.*

- The fresh charge is admitted at one end of the cylinder and the exhaust escapes at the other end. The air flow is from end to end, and little short-circuiting between the intake and exhaust openings is possible.
- The three available arrangements for uniflow scavenging are shown in Fig.8.5.



(a) (b) (c)

Fig. 8.5. Uniflow scavenging.

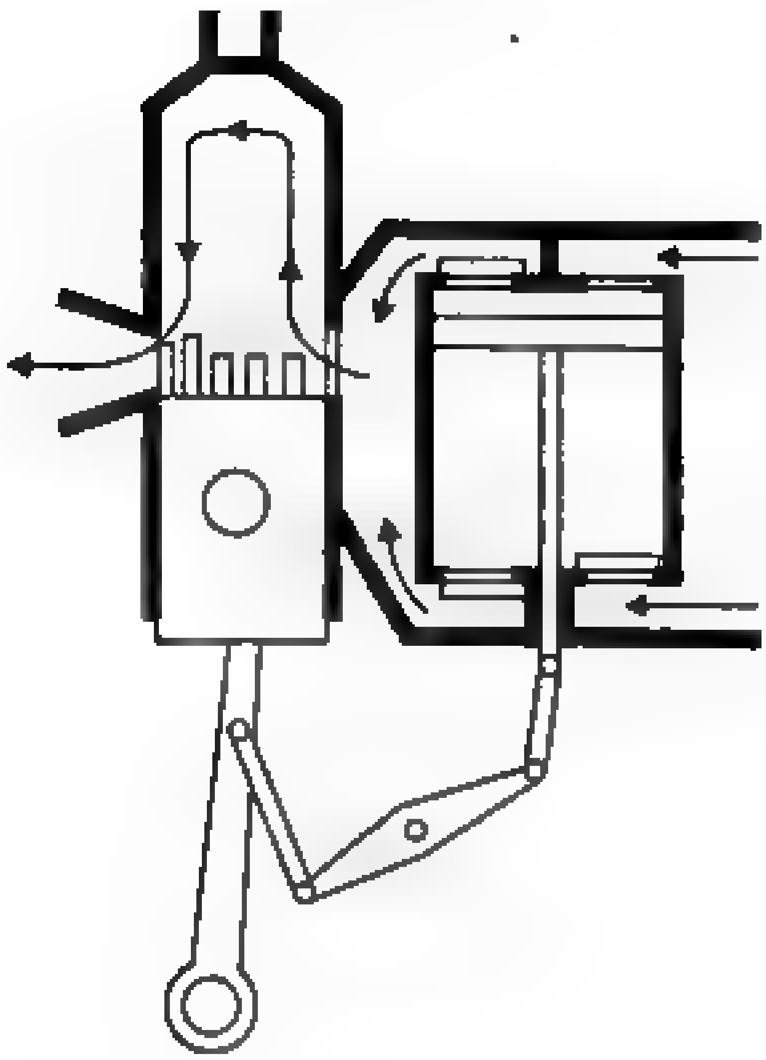


Fig. 8.9. Piston type pump.

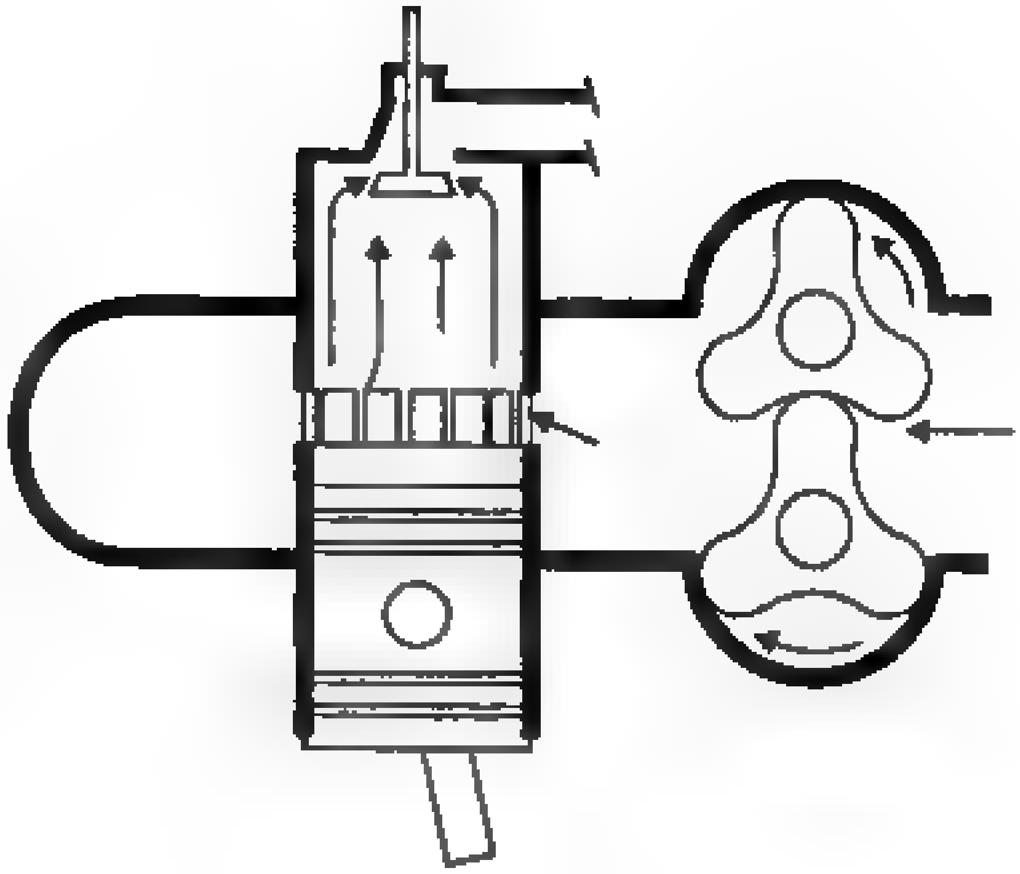


Fig. 8.10. Roots blower.

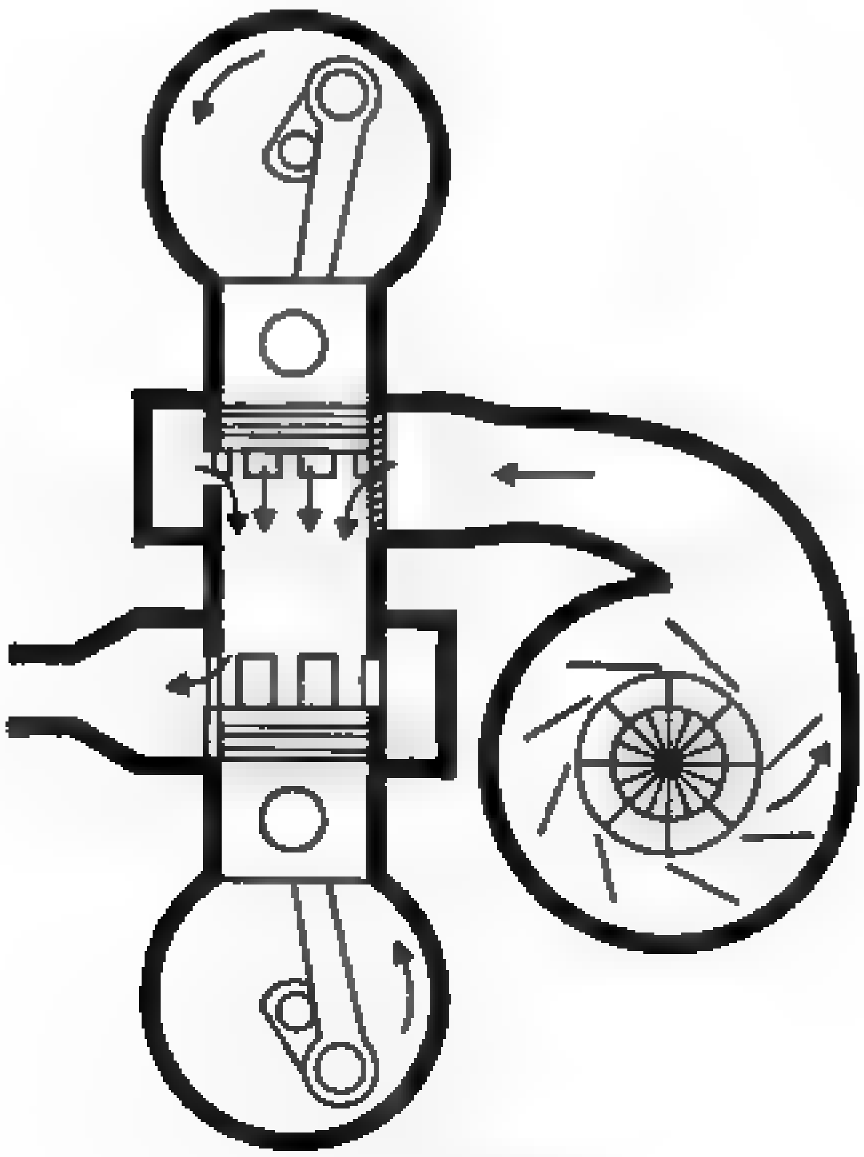


Fig. 8.11. Centrifugal blower.

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has a molecule which consists of two atoms of hydrogen and one atom of oxygen. The atoms of different elements have different masses and these values are important when a quantitative analysis is required. The actual masses are infinitesimally small, and the ratios of the masses of atoms are used. These ratios are indicated by **atomic weight** quoted on a scale which defines the atomic weight of oxygen as 16.

The symbols and molecular weight of some important elements, compounds and gases are given in the Table 9.1.

Table 9.1. Symbols and Molecular weights

Elements/Compounds/Gases	*Molecule*		*Atom*	
	Symbol	*Molecular weight*	*Symbol*	*Molecular weight*
Hydrogen	H_2	2	H	1
Oxygen	O_2	32	O	16
Nitrogen	N_2	[redacted]	N	14
Carbon	C	12	C	12
Sulphur	S	32	S	32
Water	H_2O	18	—	—
Carbon monoxide	CO	28	—	—
Carbondioxide	CO_2	[redacted]	—	—
Sulphurdioxide	SO_2	[redacted]	—	—
Marsh gas (Methane)	CH_4	16	—	—
Ethylene	C_2H_4	[redacted]	—	—
[redacted]	C_2H_6	30	—	—

9.1.3. Fuels

Fuel may be *chemical* or *nuclear*. Here we shall consider briefly *chemical fuels* only.

A chemical fuel is a substance which releases heat energy on combustion. The principal combustible elements of each fuel are *carbon* and *hydrogen*. Though *sulphur* is a combustible element too but its presence in the fuel is considered to be *undesirable.*

Fuels can be classified according to whether :

(*i*) they occur in nature called **primary fuels** or are prepared called **secondary fuels** ;

(*ii*) they are in solid, liquid or gaseous state. The detailed classification of fuels can be given in a summary form as follows :

Type of fuel	*Natural (Primary)*	*Prepared (secondary)*
Solid	Wood	Coke
	Peat	Charcoal
	Lignite coal	Briquettes
Liquid	Petroleum	Gasoline
		Kerosene
		Fuel oil
		Alcohol
		Benzol
		Shale oil

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- The relation between cetane number and delay period for a particular engine at a particular set of running conditions is illustrated in Fig 9.20.

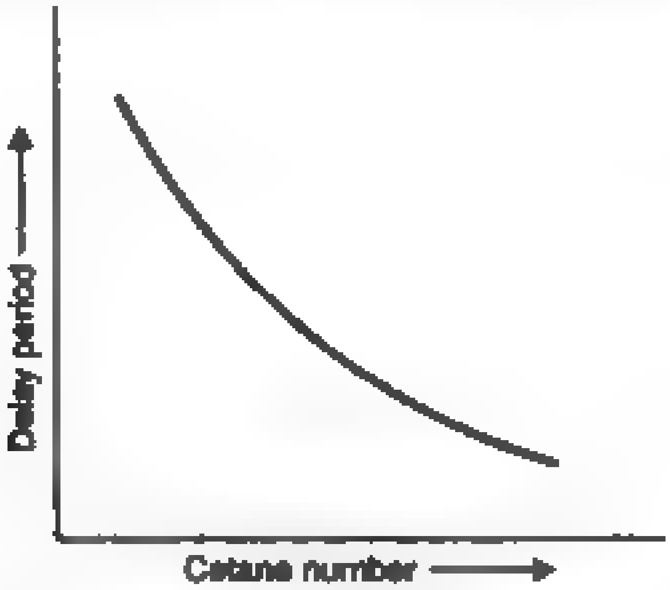

Fig. 9.20

- An approximate inverse relationship between cetane (CN) and octane (ON) number is given as :

$$CN = 60 - \frac{ON}{2}$$
(accurate within ± 5%)

9.2.10.2. Diesel Index (DI)

- The diesel index is a *cheap method of predicting ignition quality.*
- This scale is made possible because ignition quality is sensitive to hydrocarbon compositions ; that is, *paraffins have high ignition quality and aromatic and compounds have low ignition quality.*
- The *diesel index gives an indication of the ignition quality obtained from certain physical characteristics of the fuel as opposed to an actual determination in a test engine.* The index derived from the knowledge of aniline point American Petroleum Institute (API) gravity, which is put together as follows :

$$\text{Diesel index (DI)} = \text{Aniline point (°F)} \times \frac{\text{API gravity (deg)}}{100}$$

— **The aniline point of the fuel** is *the temperature at which equal parts of fuel and pure aniline dissolve in each other.* It therefore gives an indication of the chemical composition of the fuel since the more paraffinic the fuel the higher the solution temperature. Likewise, a higher API gravity reflects a low specific gravity and indicates a high paraffinic content, which corresponds to a good ignition quality.

Note. The correlation between the diesel-index and cetane number is *not exact* and *with certain fuel consumption it is not reliable but, nevertheless, it can be a useful indicator for estimating ignition quality.*

9.3. ALTERNATIVE FUELS FOR I.C. ENGINES

9.3.1. General Aspects

- The *crude oil and petroleum products,* sometimes during the 21st century will *become very scarce and costly to find and produce.* At the same time, there will likely be an increase in the number of automobiles and other I.C. engines. Although fuel economy of engines is greatly improved from the past and will probably continue to be improved,

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From table :

O_2 required per kg of coal $= 2.636$ kg

$\therefore$ Air required per kg of coal $= \dfrac{2.636}{0.233} = 11.31$ kg

(where air is assumed to contain 23.3% O_2 *by mass*)

N_2 associated with this air $= 0.767 \times 11.31 = 8.67$ kg

$\therefore$ Total N_2 in products $= 8.67 + 0.01 = 8.68$ kg

The stoichiometric A/F ratio = 11.31/1. (Ans.)

When 30 per cent excess air is used :

(i) Actual A/F ratio :

Actual A/F ratio $= 11.31 + 11.31 \times \dfrac{30}{100} = 14.7/1.$ **(Ans.)**

(ii) Wet and dry analyses of products of combustion by volume :

As per actual A/F ratio, N_2 supplied $= 0.767 \times 14.7 = 11.27$ kg

Also O_2 supplied $= 0.233 \times 14.7 = 3.42$ kg

(where air is assumed to contain $N_2 = 76.7\%$ and $O_2 = 23.3$)

In the products then, we have

$$N_2 = 11.27 + 0.01 = 11.28 \text{ kg}$$

and $$\text{excess } O_2 = 3.42 - 2.636 = 0.784 \text{ kg}$$

The products are entered in the following table and the analysis by volume is obtained :

- In column 3 the percentage by mass is given by the mass of each product divided by the total mass of 15.66 kg.
- In column 5 the moles per kg of coal are given by equation $n = \dfrac{m}{M}$. The total of column 5 gives the total moles of wet products per kg of coal, and by subtracting the moles of H_2O from this total, the total moles of dry products is obtained as 0.5008.
- Column 6 gives the proportion of each constituent of column 5 expressed as a percentage of the total moles of the wet products.
- Similarly column 7 gives the percentage by volume of the dry products.

Product	*Mass/kg coal*	*% by mass*	*M*	*Moles/kg coal*	*% by vol. wet*	*% by vol. dry*
1	*2*	*3*	*4*	*5*	*6*	*7*
CO_2	3.23	20.62	44	0.0734	14.10	14.66
H_2O	0.36	2.29	18	0.0200	3.84	—
SO_2	0.01	0.06	64	0.0002 (say)	0.04	0.04
O_2	0.78	4.98	32	0.0244	4.68	4.87
N_2	11.28	72.03	28	0.4028	77.34	80.43
	15.66 kg			Total wet = 0.5208	100.00	100.00 (Ans.)
				$- H_2O$ = 0.0200		
				Total dry = 0.5008		

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36. Give the advantages of using alternate fuels.
37. Discuss different properties of ethanol and methanol and compare them with gasoline.
38. Why blends of either ethanol or methanol are preferred over pure alcohol fuels ?
39. Give the advantages of alcohol as a fuel.
40. List the advantages of methanol as a fuel.
41. What modifications in engine are required when blends are used ?
42. State the advantages and disadvantages of hydrogen as I.C. engine fuel.
43. What is natural gas ?
44. What are the properties of CNG ?
45. What are the advantages and disadvantages of CNG ?
46. Explain briefly LPG and LNG.
47. What is Biogas ?
48. What are the properties of biogas ?

UNSOLVED EXAMPLES

1. Determine the gravimetric analysis of the products of complete combustion of acetylene (C_2H_2) with 125 per cent stoichiometric air. [Ans. CO_2 = 19.5%, H_2O = 3.9%, O_2 = 4.4%, N_2 = 72.2%]
2. One kg of ethane (C_2H_6) is burned with 80% of theoretical air. Assuming complete combustion of the hydrogen in the fuel determine the volumetric analysis of the dry products of combustion.
[Ans. CO_2 = 4.8%, CO = 11.2%, N_2 = 84%]
3. The gravimetric analysis of a sample of coal is given as 80% C, 12% H_2 and 8% ash. Calculate the stoichiometric A/F ratio and the analysis of the products by volume.
[Ans. CO_2 = 13.6%, H_2 = 12.2%, N_2 = 74.2%]
4. Calculate the stoichiometric air fuel ratio for the combustion of a sample of dry anthracite of the following composition by mass :
C = 90 per cent ; H_2 = 3 per cent , N_2 = 1 per cent ; Sulphur = 0.5 per cent ; ash = 3 per cent.
If 20 per cent excess air is supplied determine :
(i) Air fuel ratio
(ii) Wet analysis of the products of combustion by volume.
[Ans. 11.25/1 (i) 13.5/1 ; (ii) CO_2 = 16.3%, H_2O = 0.03%, SO_2 = 3.51%, N_2 = 80.3%]
5. The following is the analysis of a supply of coal gas .
H_2 = 49.4 per cent ; CO = 18 per cent ; CH_4 = 20 per cent ; C_4H_8 = 2 per cent ; O_2 = 0.4 per cent ; N_2 = 6.2 per cent ; CO_2 = 4 per cent.
(i) Calculate the stoichiometric A/F ratio.
(ii) Find also the wet and dry analysis of the products of combustion if the actual mixture is 20 per cent weak. [Ans. (i) 4.06/1 by volume ; (ii) Wet analysis : CO_2 = 9.0%, H_2O = 17.5%, O_2 = 3.08%, N_2 = 70.4%. Dry analysis : CO_2 = 10.9%, O_2 = 3.72%, N_2 = 85.4%]
6. Find the stoichiometric air fuel ratio for the combustion of ethyl alcohol (C_2H_6O), in a petrol engine. Calculate the air fuel ratios for the extreme mixture strengths of 90% and 120%. Determine also the wet and dry analysis by volume of the exhaust gas for each mixture strength.
[Ans. 8.96/1 ; 9.95/1 ; 7.47/1, Wet analysis : CO_2 = 11.2%, H_2O = 16.8%, O_2 = 1.85%, N_2 = 70.2%
Dry analysis : CO_2 = 13.45%, O_2 = 2.22%, N_2 = 84.4%
Wet analysis : CO_2 = 6.94%, CO = 6.94%, H_2 = 20.8%, N_2 = 65.3%
Dry analysis : CO_2 = 8.7%, CO = 8.7%, N_2 = 82.5%]
7. For the stoichiometric mixture of Example 7 10 calculate :
(i) The volume of the mixture per kg of fuel at a temperature of 65°C and a pressure of 1.013 bar.
(ii) The volume of the products of combustion per kg of fuel after cooling to a temperature of 120°C at a pressure of 1 bar. [Ans. (i) 9.226 m^3 ; (ii) 11.58 m^3]

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- For a S.I. engine, the design of carburetion system is very complicated owing to the fact that the air-fuel ratio required by it varies widely over its range of operation, particularly for an automotive engine. For idling as well as for maximum power rich mixture is required.

11.4. MIXTURE REQUIREMENTS

Fig. 11.2. shows the variation of mixture requirements from no-load to full-load in a S.I. engine.

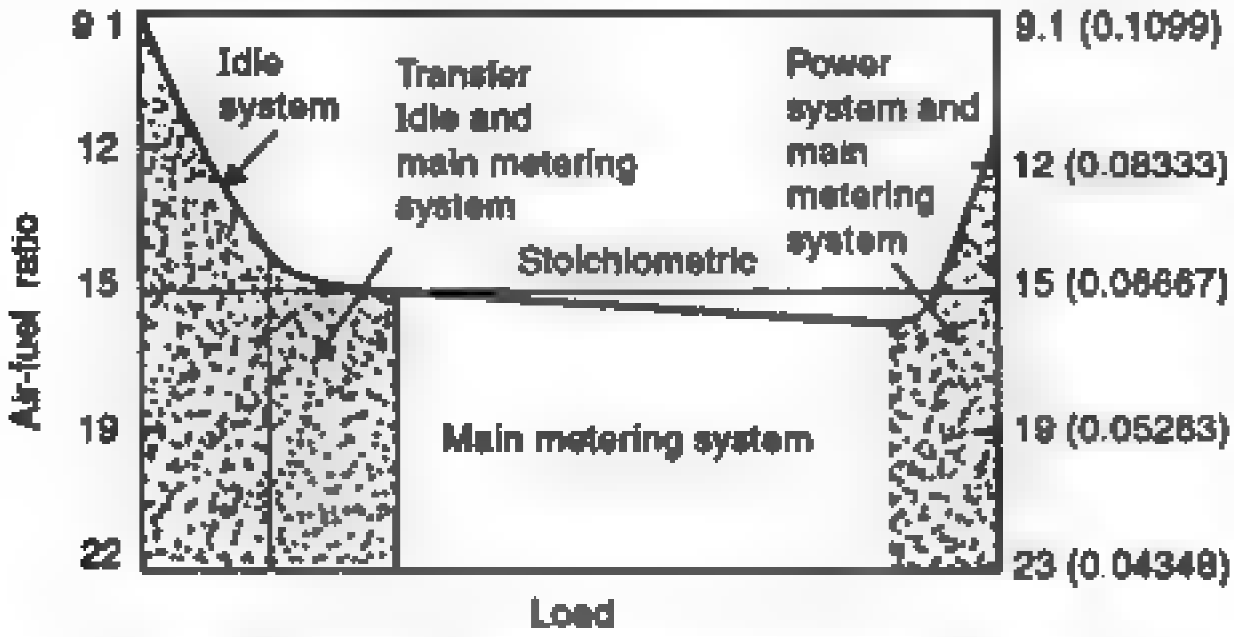

Fig. 11.2. Mixture requirements of automotive S.I. engine.

1. Idling and low speed (*From no-load to about 20% of rated power*) :

Idling refers to *no power demand.* During idling air supply is throttled and residual gases make up a large fraction of the charge at the end of the suction period. In addition, during valve overlap period some exhaust gases are drawn back into the cylinder. The result is that a chemically correct (stoichiometric) mixture of air and fuel (= 15 : 1) would be so diluted by residual gases that combustion would be erratic or impossible. *A rich mixture, therefore, must be supplied during idling (say A/F ratio* 11 : 1 or 12 : 1). *The richness should gradually change to slightly lean for the second range as* shown in Fig. 11.2.

2. Cruising or normal power (*from about 25% to about 75% of rated power*) :

In the *normal power range the main consideration is fuel economy.* Because mixture of fuel and air is never completely homogeneous the stoichiometric mixture of fuel and air will not burn completely and some fuel will be wasted. For this reason an *excess of air, say 10% above theoretically correct (= 16.5 : 1), is supplied in order to ensure complete burning of the fuel.*

3. Maximum power (*From 75% to 100% of rated power*):

Maximum power is obtained when all the air supplied is fully utilized. As the mixture is not completely homogeneous a rich mixture must be supplied to assure utilization of air (though this would mean wasting some fuel, which would pass in exhaust in unburned state). The *air-fuel ratio for maximum power is about 13 : 1.*

Running on the weakest mixture. *This results in high efficiency and there is fuel economy. On normal loads engines work on weak mixture.*

Running on richest mixture. *Engines run on rich mixture during idling and during the overload. The effect is lowering of efficiency and pollution problems.*

- Automobiles engines generally operate well below full power and a complicated system for making the mixture rich is neither called for nor economically advisable, although some means are employed to enrich the mixture. A more representative curve for an automobile engine is shown in Fig 11.3.

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- It consists of a *float chamber, nozzle with metering orifice, venturi* and *throttle valve.*
  — The **float chamber** is meant for storage of fuel. The fuel supplied under gravity action or by fuel pump enters the float chamber through a filter. The arrangement is such that when oil reaches a particular level the *needle/float valve* blocks the inlet passage and thus cuts off the fuel oil supply. On the fall of oil level, the *float* descends down, consequently intake passage open and again the chamber is filled with oil. Then the float and the needle/float valve maintains a constant fuel oil level in the float chamber. There is a *nozzle (discharge jet)* from which the fuel is sprayed into the air stream as it enters the inlet and passes through the venturi or throat. *The fuel level is slightly below the outlet of the jet when the carburettor is inoperative.*
- As the piston moves down in the engine cylinder, suction is produced in the cylinder as well as in the induction manifold as a result of which air flows through the carburettor. *The velocity of air increases as it passes through the constriction at the venturi and the pressure decreases due to conversion of a portion of pressure head into kinetic energy.* Due to decreased pressure at the venturi and hence by virtue of difference in pressure (between the float chamber and the venturi) the jet issues fuel oil into air stream. Since the jet has a fine bore, the oil issuing from the jet is in the form *of fine spray* ; it vaporises quickly and mixes with air. The air-fuel mixture enters the engine cylinder ; *it quantity being controlled by the position of the "throttle valve".*

Limitations :

(*i*) Although theoretically the air-fuel ratio supplied by a simple (single jet) carburettor should remain constant as the throttle goes on opening, actually *it provides increasingly richer mixture as the throttle is opened.* This is because of the reason that the density of air tends to decrease as the rate of flow increases.

— This fault is corrected by *using a number of compensating devices.*

(*ii*) During idling, however, the nearly closed throttle causes a reduction in the mass of air flowing through the venturi. At such low rates of air flow, the pressure difference between the float chamber and the fuel discharge nozzle becomes very small. It is not adequate enough to cause fuel to flow through the jet.

— This fault may be corrected by *using an idling jet which helps, in running the engine during idling.*

(*iii*) Carburettor does not have arrangement for providing rich mixture during starting and warm up.

— This limitation is taken of by using a *choke arrangement.*

11.8. COMPLETE CARBURETTOR

For meeting the demand of the engine under all conditions of operation, the following additional devices/systems are added to the simple carburettor :

1. Main metering system
2. Idling system
3. Power enrichment or economiser system
4. Acceleration pump system
5. Choke.

1. Main metering system :

The main metering system of a carburettor should be so designed as to supply a *nearly constant fuel-air ratio over a wide range of operation.* This F/A ratio is approximately equal to 0.064 (A/F ratio = 15.6) for best economy at full throttle. In order to correct the tendency of the simple carbu-

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orifice is $(p_1 - p_2)$ when p_2 is the pressure at the throat. If the valve is closed, the float chamber communicates only with venturi throat and pressure on the fuel surface will be p_2. Then the carburettor depression Δp will be zero and no fuel can flow. By proper adjustment of control valve any pressure between p_1 and p_2 can be obtained in the float chamber, thus altering the quantity of fuel discharged by the nozzle.

(*iv*) ***Auxiliary valve carburettor :***

Fig. 11.10 shows an auxiliary valve carburettor. When load on the engine increases, the vacuum at the venturi throat also increases. This lifts the valve against the spring force and *consequently more air is admitted and the mixture is prevented from becoming over-rich.*

(*v*) ***Auxiliary port carburettor.***

- This method is used in *aircraft carburettors for altitude compensation.*
- Fig. 11.11 shows an auxiliary port carburettor. When the butterfly valve is opened, additional air is admitted and at the same time the depression at the venturi throat is reduced ; this results in decreasing the quantity of fuel drawn in.

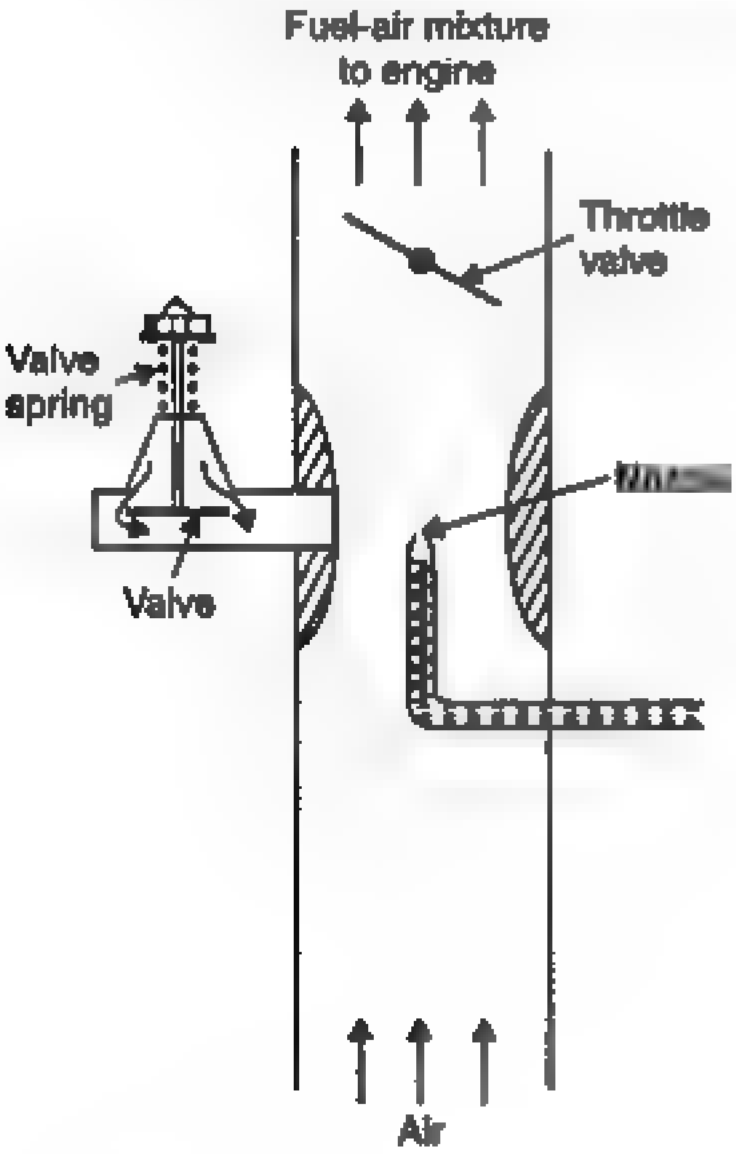

Fig. 11.10. An auxiliary valve carburettor.

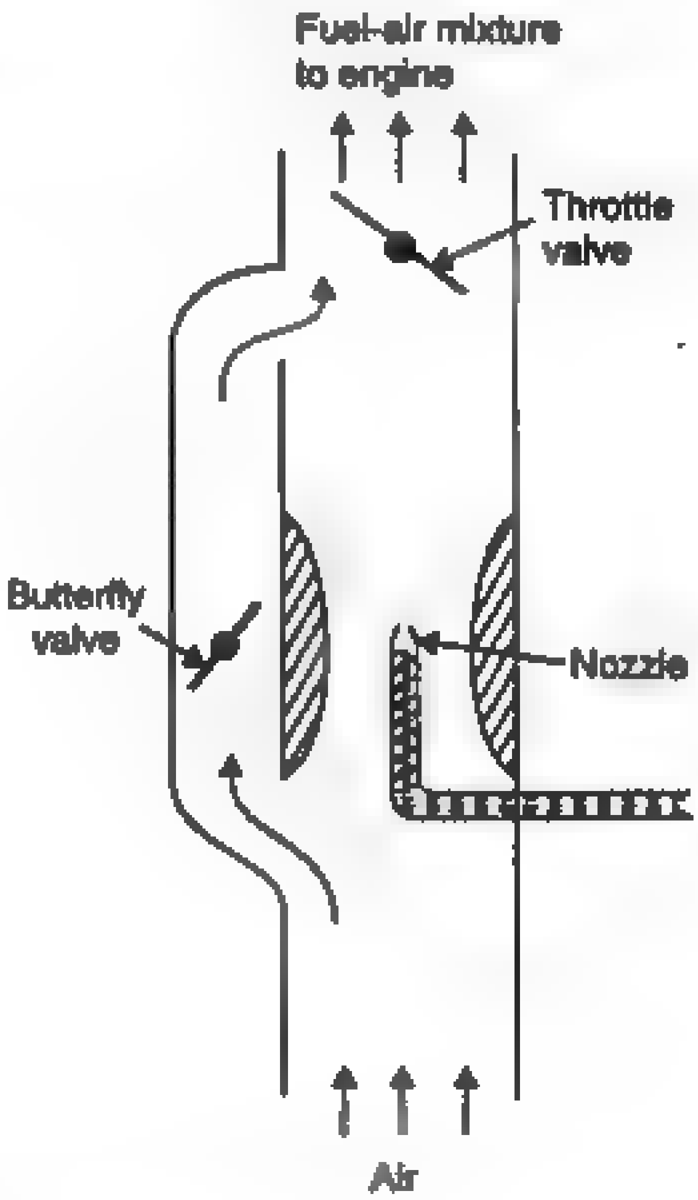

Fig. 11.11. An auxiliary port carburettor.

2. Idling system :

- As earlier discussed that at idling and low load an engine requires a rich mixture having about air-fuel ratio 12 : 1. The main metering system not only fails to supply enrich the mixture at low air flows but also cannot supply any fuel during idling operation. It is due to this reason that a separate idling jet must be incorporated in the basic carburettor.

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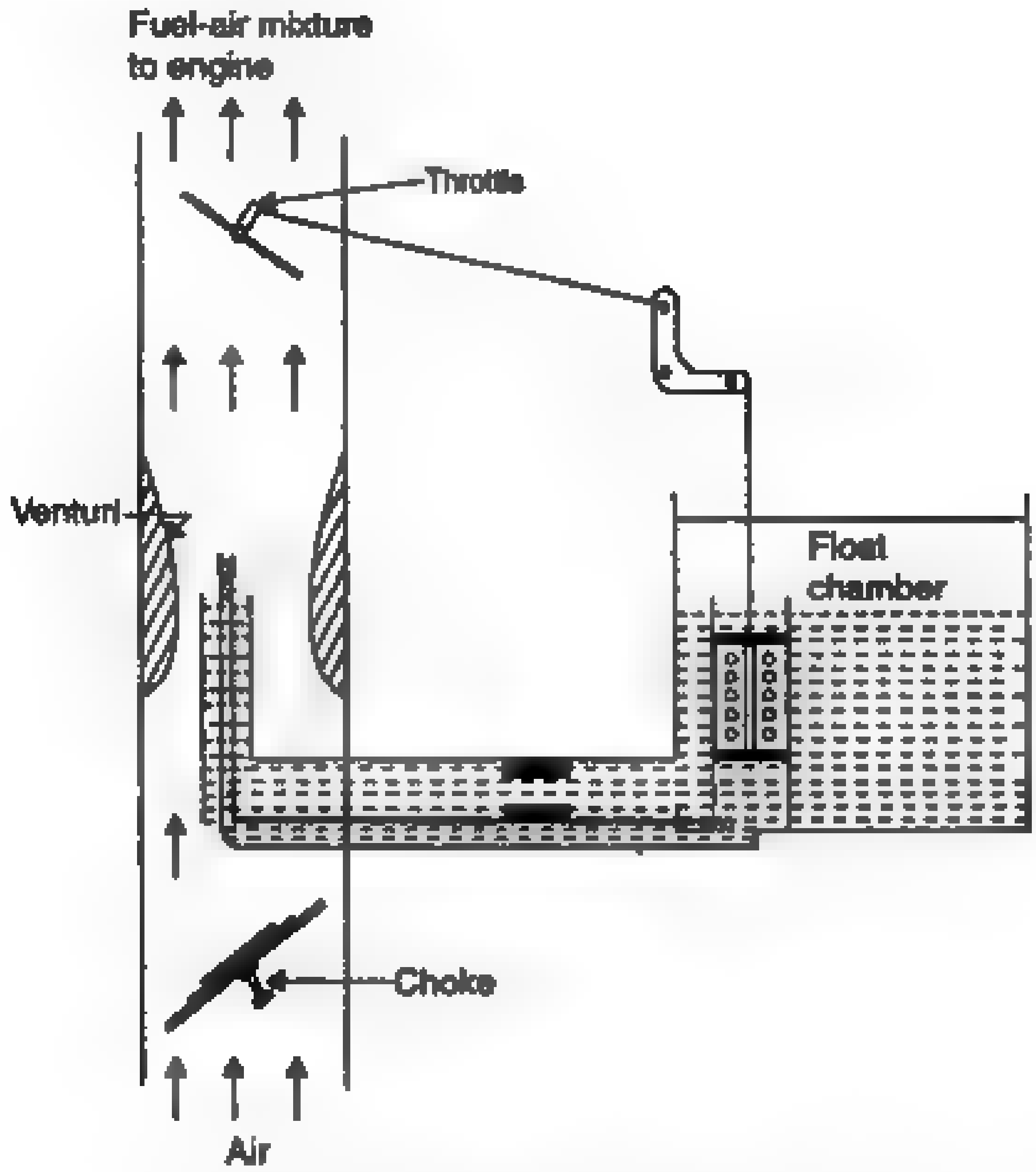

Fig. 11.15. Choke valve with spring-loaded by-pass.

- The provision of *auxiliary fuel jets* that are opened manually or automatically only as required, is an *alternative to the choke.*

11.9. CARBURETTORS

11.9.1. Essential Features of Good Commercial Carburettor for Automotive Engines

Carburettor is a *mixing device to supply the engine with air-fuel mixture. It atomizes the fuel and mixes it with air in varying proportions to meet the changing operating conditions of automotive engines.* It is required to provide the following *essential features* :

1. To meter and supply the proper quantity and proportion of air and fuel at correct strength under all conditions of load and speed of the engine of the car for

(*i*) starting it easily from cold.

(*ii*) providing a rich mixture for slow idling.

(*iii*) providing a rich mixture for acceleration,

(*iv*) providing a rich mixture for high speed, and

(*v*) providing a rich mixture for low speed when moving up-gradient.

2. To operate satisfactorily when cold, or when hot

3. To operate satisfactorily both on level and hills

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applied to the starting passage (11), sucking petrol from jet (9) and air from jet (10). The jets and passages are so shaped that the mixture provided to the carburettor is rich enough for starting.

- After starting the engine, the starter lever is brought to the intermediate position, bringing the smaller holes in the starter valve (8) into the circuit, thus reducing the amount of petrol. Also in this position, the throttle valve is partly open, so that the petrol is also coming from the main jet. In this situation, the reduced mixture supply from the starter system, however, is sufficient to keep the engine running till it reaches the normal running temperature, when the starter is brought to *"off-position"*.

3. Idling and slow running :

- From the lower part of the well of the emulsion system a hole leads off to the pilot jet (13).
- At idling the throttle is practically closed and therefore the suction created by the engine on suction stroke gets communicated to the pilot jet. Fuel is inducted therefrom, and mixed with a small amount of air admitted through the small pilot air bleed orifice (14) and forms an emulsion which is conveyed down the vertical channel and discharged into the throttle body past the idling volume control screw (15). The *slow running adjustment screws allows the engine speed to be varied.*
- *By-pass orifice* (17) provided on the venturi side of the throttle valve *ensures the smooth transfer from idle and low speed circuit to the main jet circuit without occurrence of flat spot.*

4. Acceleration :

- In order to avoid flat spot during acceleration, a diaphragm type *acceleration* is incorporated (also known as economy system). This pump supplies spurts of extra fuel needed for acceleration through pump injector (18).
- Pump lever (19) is connected to the accelerator so that on pressing the pedal, the lever moves towards left, pressing the membrane towards left, thus forcing the petrol through pump jet (20) and injector (18). On making the pedal free, the lever moves the diaphragm back towards right creating vacuum towards left which opens the pump inlet valve (21) and thus admits the petrol from the chamber into pump.

11.9.3.2. Carter Carburettor

A *carter carburettor* is an American make carburettor and is used in *jeeps*. It is a *standard equipment on chevorolet and Pontiac series of cars.*

Fig. 11.18 shows the schematic arrangement of a downdraft type Carter carburettor. The brief description of the components and circuits is given below :

- The petrol (fuel) enters the conventional type float chamber (1).
- The air enters the carburettor from the top, a choke valve (2) in the passage remains open during normal working.
- This carburettor has a triple venturi diffusing type of choke, *i.e.* it has *three venturies*, the smallest (3) lies above the level in the float chamber, and the remaining two venturies (6) and (5) are below the fuel level (in the float chamber), one below the other.
- At very low speeds, suction in primary venturi (3) is sufficient to draw the fuel. The nozzle (4) enters the primary venturi at an angle, and throws the fuel up against the air stream evenly, thereby providing finely divided atomised fuel. The mixture from venturi (3) passes centrally through the secondary venturi (5) where it is surrounded by a blanket of air stream and finally this leads to the third main venturi (6), where again the fresh air supply insulates the stream from the secondary venturi. The fuel-air mixture

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2. **Mixture adjustment and fuel temperature compensation :**

- The jet height initial adjustment and hence mixture strength can be made by altering the tilt of the *right angled lever* which is attached to a *spring-loaded retaining screw* and a bimetallic strip which extends to the petrol jet. To alter the jet height, the horizontal jet adjustment screw is screwed inwards to lower the jet and enrich the mixture, and outwards to raise the jet and weaken the mixture.
- In order to compensate for the variation in fuel viscosity within changing temperature and the reluctance of the fuel to flow through an orifice as its viscosity rises, a **bimetallic strip** *submerged in the fuel senses a temperature change and alters the effective jet size accordingly.* When the fuel temperature rises the bimetallic strip curls upwards and pushes the jet further into the tapered needle. Conversely, if the fuel becomes cooler, the strip bends downwards and lowers the jet to increase the annular jet orifice.

3. **Part throttle by pass emulsion system.** Refers Fig. 11.19 (*a*).

- This system is a passagaway which bypasses the mixing chamber, it spans the distance between the feed duct at the jet bridge and a discharge duct at the throttle butterfly edge.
- The bypass passage, with a small throttle opening, delivers a quantity of mixture in a well emulsified condition from the jet to a high depression point near the edge of the throttle. Since the bypass passageway is much smaller than the mixing chamber bore, the mixture velocity through this passage will be much greater and therefore the air-fuel mixing will be that much more thorough.

4. **Overrun valve.** Refer Fig. 11.19 (*b*).

- Under *overrun working conditions,* the closed throttle will create a very high depression on the engine side of the throttle and in the induction manifold. Consequently, the *effective compression ratio will be low, burning will be slow and erratic, and the exhaust products will contain high valves of hydrocarbon. To improve the burning process so that more of the fuel is doing useful work* and less is passed through to the exhaust as incomplete combustion products, a **spring-loaded plate-valve** is *incorporated in the throttle butterfly disc.*
- When the engine is operating at overrun conditions, the manifold depression at some predetermined value will force open the spring loaded plate-valve to emit an additional quantity of correct air-fuel mixture. *The increased supply of air-fuel mixture will reduce the manifold depression with the result that the denser and better prepared mixture charge will improve combustion, and hence less unburnt products will be passed through to the exhaust.*

5. **Hydraulic damper (acceleration enrichment device).** Refers Fig. 11.19 (*c*).

- This device is incorporated *to enrich the mixture strength when the throttle is opened rapidly but it does not interfere with the normal air-valve lift or fall as the mixing chamber depression changes with respect to steady throttle opening.*
  — The damper valve is mounted on the lower end of a long stem inside the hollow guide spindle of the air-valve and is submerged in a light oil. The damper consists of a vertically positioned loose fitting sleeve, its underside resting on a spring clip attached to stem, while its upper end is chamfered so that it matches a conical seat formed on the central support stem.
- On rapid opening of the throttle, the sudden rise in depression in both mixing chamber and air valve upper chamber tends to jerk up the air valve assembly. Simultaneously, the viscous drag of oil in the hollow spindle will lift the sleeve and press it against its seat, and so the oil is thus temporarily trapped beneath the damper so that it prevents

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A major feature *with petrol injection is that there is separate air and fuel metering and that fuel metering is precise under all engine operating condition's.*

11.10.5. Injection Considerations

The fuel can be discharged into the air stream, *using indirect injection arrangements,* by the following *two methods* :

1. Continuous injection.
2. Intermittent or pulsed injection.

1. Continuous injection :

In this arrangement, *the injector nozzle and valve are permanently open* while the engine is operating and the *amount of fuel discharged in the form of a spray is controlled by either varying the metering orifice or the fuel discharge pressure, or a combination of both of these possible variables.*

2. Intermittent or pulsed injection :

In this type of injection, *fuel is delivered from the injector in spray form at regular intervals with a constant fuel discharge pressure and the amount of fuel discharged is controlled by the time period the injector nozzle valve is open.*

- ***Timed injection.*** This where the start of delivery for each cylinder *occurs at the same angular point in the engine cycle,* this can be anything from 60° to 90° after T.D.C. on the induction stroke.
- ***Non-timed injection.*** In contrast to timed injection, this is where *all the injectors are programmed to discharge their spray at the same time, therefore each cylinder piston will be on a different part of the engine cycle.*

11.10.6. Comparison of Petrol Injection and Carburetted Fuel Supply Systems

Merits of petrol injection :

Following are the *merits of petrol engine system :*

1. In petrol injection system, *due to absence of venturi there is the minimum of air restriction so that higher engine volumetric efficiencies can be obtained with the corresponding improvement in power and torque.*

2. *The spots* for pre-heating the cold air and fuel mixture are *eliminated* so that *denser air enters the cylinder* when the engine has reached normal operating conditions.

3. As the manifold branch pipes are not greatly concerned with mixture preparation they *can be designed to utilize the inertia of the air charge to increase the engine's volumetric efficiency ;* (this does not apply for single point injection).

4. Because of *direct spray discharge* into each inlet port, *acceleration response is better.*

5. *Atomization of fuel droplets is generally improved* over normal speed and load driving conditons.

6. It is possible to *use greater inlet and exhaust valve overlap without poor idling, loss of fuel or increased exhaust pollution.*

7. The monitoring of engine operating parameters enables accurate matching of air and fuel requirements under normal speed and load conditions which *improves engine performance, fuel consumption and reduces exhaust pollution.*

8. Fuel injection equipment is *precise in metering injected fuel spray* into the intake ports over the complete engine speed, load and temperature operating range.

9. There is *precise fuel distribution* between engine cylinders even under full load conditions *with multi-point injection.*

10. *Multi-point injection does not require time for fuel transportation in the intake manifold and there is no manifold wall melting.*

11. With fuel injection, when cornering fast or due to heavy braking, fuel surge is eliminated.

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where, ρ_f = Constant density of fuel, and

C_f = Velocity of flow of fuel.

$$\therefore \qquad C_f = \sqrt{\frac{2(p_1 - p_2 - gz\rho_f)}{\rho_f}} = \sqrt{\frac{2(\Delta p_a - gz\rho_f)}{\rho_f}} \qquad ...(11.6)$$

[It may be noted that due to petrol surface being lower than the top of the jet by z metres the pressure difference becomes $(\Delta p_a - gz\rho_f)$ instead of Δp_a]

Mass of fuel per second, m_f (theoretical) $= A_f \rho_f = A_f \sqrt{2\rho_f(\Delta p_a - gz\rho_f)}$ ...(11.7)

where, A_f = Cross-sectional area of the fuel jet m^2.

$\therefore$ Air-fuel (A/F) ratio,

$$\frac{m_a}{m_f} = \frac{A_2 \sqrt{2\rho_a \, \Delta p_a}}{A_f \sqrt{2\rho_f \, (\Delta p_a - gz\rho_f)}} = \frac{A_2}{A_f} \sqrt{\frac{\rho_a}{\rho_f}} \cdot \sqrt{\frac{\Delta p_a}{(\Delta p_a - gz\rho_f)}} \qquad ...(11.8)$$

If C_{da} and C_{df} are the coefficients of discharge of venturi and fuel jet respectively, then

$$\frac{\dot{m}_a}{m_f} = \frac{C_{da}}{C_{df}} \cdot \frac{A_2}{A_f} \cdot \sqrt{\frac{\rho_a}{\rho_f}} \cdot \sqrt{\frac{\Delta p_a}{\Delta p_a - gz\rho_f}} \qquad ...(11.9)$$

If $z = 0$,

$$\frac{\dot{m}_a}{m_f} = \frac{C_{da}}{C_{df}} \cdot \frac{A_2}{A_f} \sqrt{\frac{\rho_a}{\rho_f}} \qquad ...(11.10)$$

Case II. *Taking into consideration the compressibility of air in account*—**Exact Analysis.**

When the compressibility of air is taken into account, the air flow will change but the fuel flow will remain unchanged. Applying steady flow energy equation (S.F.E.E.) at sections 1-1 and 2-2, we get,

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W$$

or

$$Q - W = (h_2 - h_1) + \frac{C_2^2 - C_1^2}{2}$$

where h_1, h_2 = Enthalpies at sections 1-1 and 2-2 respectively.

Since $Q = 0$, $W = 0$ and $C_1 = 0$

$$\therefore \qquad C_2 = \sqrt{2(h_1 - h_2)}$$

Substituting $h_1 = c_p T_1$ and $h_2 = c_p T_2$, we get

$$C_2 = \sqrt{2c_p T_1 \left(1 - \frac{T_2}{T_1}\right)} \qquad ...(11.11)$$

Since the flow process between the atmosphere and the venturi throat is isentropic,

$$\therefore \qquad \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \qquad ...(11.12)$$

Substituting eqn. (11.6) in eqn. (11.5), we get

$$C_2 = \sqrt{2c_p T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma}\right]} \qquad ...(11.13)$$

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$$\frac{7.2}{3600} = \frac{\pi}{4}(d_f)^2 \times 0.7 \ \sqrt{2 \times 750\,(2922 - 9.81 \times 0.0042 \times 750)}$$

$$= 1144.89\,(d_f)^2$$

$$\therefore \qquad \mathbf{d_f} = \left(\frac{7.2}{3600 \times 1144.89}\right)^{1/2} = 1.32 \times 10^{-3} \text{ m} \quad \text{or} \quad \mathbf{1.32 \text{ mm.}} \quad \textbf{(Ans.)}$$

Example 11.4. *A simple carburettor under a certain condition delivers 5.45 kg/h of petrol with an air-fuel ratio of 15. The fuel jet area is 2 mm² with a coefficient of discharge of 0.75. If the tip of the fuel jet is 0.635 cm above the level of petrol in the float chamber and the venturi throat coefficient of discharge is assumed to be 0.80, calculate :*

(i) The venturi depression in cm of H_2O necessary to cause air and fuel flow at the desired rate.

(ii) The venturi throat diameter.

(iii) The velocity of air across the venturi throat.

You may take density of air = 1.29 kg/m³ and specific gravity of period = 0.72.

(Madras University)

Solution. *Given :* $\dot{m}_f = \frac{5.45}{3600} = 0.001514$ kg/s ; A/F ratio = 15 ; $A_f = 2$ mm² $= 2 \times 10^{-6}$ m² ; $C_{df} = 0.75$; $z = 0.635$ cm $= 0.00635$ m ; $C_{da} = 0.8$;
$\rho_a = 1.29$ kg/m³ ; Sp. gr. of petrol = 0.72.

(i) Venturi depression, Δp_a :

$$\dot{m}_f = C_{df} \cdot A_f \ \sqrt{2\rho_f(\Delta p_a - gz\rho_f)} \qquad \text{...[Eqn. (11.7)]}$$

where Δp_a is in N/m².

$$0.001514 = 0.75 \times 2 \times 10^{-6} \ \sqrt{2 \times (0.72 \times 1000)\,(\Delta p_a - 9.81 \times 0.00635 \times 0.72 \times 1000)}$$

$$\frac{0.001514}{0.75 \times 2 \times 10^{-6}} = 37.95\sqrt{(\Delta p_a - 44.85)}$$

or
$$\Delta p_a - 44.85 = \left(\frac{0.001514}{0.75 \times 10^{-6} \times 37.95}\right)^2 = 707.37$$

$$\therefore \qquad \Delta p_a = 752.22 \text{ N/m}^2 = \frac{752.22}{9810} \text{ m of water} = \mathbf{7.67 \text{ cm of } H_2O.} \quad \textbf{(Ans.)}$$

(ii) Venturi throat diameter, D_t :

Air flow rate $= \frac{5.45}{3600} \times 15 = 0.02271$ kg/s

Also,
$$\dot{m}_a = C_{da}\,A_t\,\sqrt{2\rho_a \Delta p_a}$$

$$\therefore \qquad 0.02271 = 0.8 \times A_t\,\sqrt{2 \times 1.29 \times 752.22}$$

or
$$A_t = 6.444 \times 10^{-4} \text{ m}^2 = \frac{\pi}{4} D_t^2$$

$$\therefore \qquad \mathbf{D_t} = \left[\frac{6.444 \times 10^{-4} \times 4}{\pi}\right]^{1/2} = 0.0286 \text{ m} = \mathbf{2.86 \text{ cm.}} \quad \textbf{(Ans.)}$$

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Now, $$\dot{m}_a = \frac{A_2 C_2}{v_2}$$

$\therefore$ Throat area, $$A_2 = \frac{\dot{m}_a \times v_2}{C_2} = \frac{0.1 \times 0.898}{92} = 9.76 \times 10^{-4} \text{ m}^2 = 9.76 \text{ cm}^2$$

But $$A_2 = \frac{\pi}{4} D_2^2 = 9.26$$

$\therefore$ $$D_2 \ (\text{or } D_t) = \sqrt{\frac{9.76 \times 4}{\pi}} = 3.525 \text{ cm.} \quad \textbf{(Ans.)}$$

(*ii*) **Orifice diameter, d_f :**

Pressure drop at venturi = 1.013 – 0.937 = 0.076 bar

Pressure drop at jet = 0.75 × 0.076 = 0.057 bar

Now, $$\dot{m}_f = A_f C_{df} \sqrt{2\rho_f \Delta p}$$

$$0.0075 = A_f \times 0.6 \sqrt{2 \times 740 \times 0.057 \times 10^5} = 1742.68$$

$\therefore$ $$A_f = 4.304 \times 10^{-6} \text{ m}^2 \quad \text{or} \quad 4.304 \text{ mm}^2$$

But, $$A_f = \frac{\pi}{4} d_f^2 = 4.304$$

$\therefore$ $$d_f = \sqrt{\frac{4 \times 4.304}{\pi}} = 2.34 \text{ mm.} \quad \textbf{(Ans.)}$$

Example 11.8. *The following data relate to a 4-stroke petrol engine of Hindustan Ambassador :*

Capacity of the petrol engine	*= 1489 c.c.*
Speed at which maximum power is developed	*= 4200 r.p.m.*
The volumetric efficiency (at the above speed)	*= 75 percent*
The air-fuel ratio	*= 13 : 1*
Theoretical air speed at choke (at peak power)	*= 85 m/s*
The co-efficient of discharge for venturi	*= 0.82*
The co-efficient of discharge of the main petrol jet	*= 0.65*
The specific gravity of petrol	*= 0.74*
Level of petrol surface below the choke	*= 6 mm*
Atmospheric pressure and temperature	*= 1.013 bar, 20°C respectively*

An allowance should be made for the emulsion tube, the diameter of which can be taken as 40 percent of the choke diameter.

Calculate the sizes of a suitable choke and main jet.

Solution. *Given :* V_s = 1489 c.c. = 1489 × 10^{-6} m³ = 0.001489 m³ ; N = 4200 r.p.m ; $\eta_{vol.}$ = 75% ; A/F ratio = 13 : 1 ; C_t (= C_2) = 85 m/s ; C_{da} = 0.82 ; C_{df} = 0.65 ; ρ_f = 0.74 × 1000 = 740 kg/m³ ; p_1 (= p_a) = 1.013 bar ; p_2 (= p_t) = ? ; T_1 (= T_a) = 20 + 273 = 293 K ; z = 6 mm = 0.006 m ; d = 0.4 D.

$$\text{Volume of air induced} = \eta_{vol.} \times V_s$$
$$= \frac{0.75 \times 0.001489 \times 4200}{2 \times 60} = 0.03909 \text{ m}^3/\text{s}$$

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UNSOLVED EXAMPLES

1. A four-cylinder four-stroke engine having diameter and length of stroke as 100 mm and 120 mm respectively is running at 2000 r.p.m. Its carburettor venturi has a 30 mm throat. Assuming coefficient of air flow 0.8, density of air 1.2 kg/m^3 and volumetric efficiency of the engine as 70 percent, determine the suction at the throat. **[Ans.** 0.0339 bar]
2. A simple jet carburettor is required to supply 6 kg of air per minute and 0.45 kg of fuel of density 740 kg/m^3. The air is initially at 1.013 bar and 27°C. Calculate the throat diameter of the choke for a flow velocity of 91 m/s. Velocity coefficient = 0.8.

   If the pressure drop across the fuel metering orifice is 0.75 of that at the choke, calculate orifice diameter assuming C_d = 0.6. **[Ans.** 35.25 mm ; 2.34 mm]
3. A 4-stroke petrol engine of Hindustan Ambassador has a capacity of 1489 c.c. It develops maximum power at 4200 r.p.m. The volumetric efficiency at this speed is 70 percent and the air/fuel ratio is 13 : 1. At peak power the theoretical air speed at choke is 90 m/s. The coefficient of discharge for venturi is 0.85 and that of the main petrol jet is 0.66. An allowance should be made for the emulsion tube, the diameter of which can be taken as 1/2.5 of the choke diameter. The petrol surface is 6 mm below the choke at this engine condition. The specific gravity of petrol is 0.74. Atmospheric pressure and temperature are 1.013 bar and 20°C respectively.

   Calculate the sizes of a suitable choke and main jet. **[Ans.** 27.35 mm ; 1.58 mm]
4. A single jet simple carburettor is to supply 6.11 kg/min. of air and 0.408 kg/min of petrol, density 768 kg/m^3. The air is initially at 1.027 bar and 15.5°C. Calculate the throat diameter of the venturi throat if the speed of air is 97.5 m/s, assuming a velocity coefficient of 0.84. Assume adiabatic expansion and γ for air as 1.4. If the drop across fuel metering orifice be 0.8 of the pressure at the throat ; calculate the orifice diameter assuming a coefficient as 0.66. **[Ans.** 2.05 mm]
5. An engine having a simple single jet carburettor consumes 6.5 kg of fuel/hour. The fuel density is 700 kg/m^3. The level of fuel in the float chamber is 3 mm below the top of the jet when the engine is not running. Ambient conditions are 1.01325 bar and 17°C. The jet diameter is 1.25 mm and its discharge coefficient is 0.6. The discharge coefficient of air is 0.85. Air-fuel ratio is 15. Determine the critical air velocity and the throat diameter (effective). Express the pressure depression in cm of water. Neglect compressibility of air. **[Ans.** 4.945 m/s ; 19.9 mm ; 43.99 cm]
6. An eight-cylinder 4-stroke petrol engine with bore and stroke of 100 mm each uses volatile fuel of composition C = 84%, H_2 = 16%. The throat diameter of choke tube is 40 mm. The volumetric efficiency at 3000 r.p.m. is 75 percent referred to 0°C and 1.01325 bar. The pressure depression is 0.116 bar and the temperature at throat is 16°C. If chemically correct A / F ratio is supplied for consumption, determine :
   (*i*) Fuel consumption in kg / h ;
   (*ii*) The air velocity through the tube.

   Take characteristic gas-constant R for air and fuel as 287 J/kg K and 97 J/kg K respectively.

   [Ans. 35.1 kg/h ; 116 m/s]
7. The venturi of a simple carburettor has a throat diameter of 20 mm and the coefficient of air flow is 0.85. The fuel orifice has a diameter of 1.25 mm and the coefficient of fuel flow is 0.66. The petrol surface is 5 mm below the throat. Assuming density of and fuel as 1.2 kg/m^3 and 750 kg/m^3 respectively, calculate :
   (*i*) The A/F ratio for a pressure drop of 0.07 bar when the nozzle lip is neglected ;
   (*ii*) The A/F ratio when the nozzle lip is taken into account ;
   (*iii*) The minimum velocity of air or critical air velocity required to start the fuel flow when nozzle lip is provided. **[Ans.** (*i*) 13.2 , (*ii*) 13.235 ; (*iii*) 7.83 m/s]
8. A carburettor with float chamber vented to atmosphere is tested in a laboratory without the air cleaner. The A / F ratio as calculated is 15 at the atmospheric conditions of 1.008 bar. The pressure recorded at the throat is 0.812 bar.

   This carburettor is fitted with air cleaner and once again tested. The additional pressure drop due to air cleaner is 0.04 bar with the air flow at the atmospheric conditions to remain unchanged at 260 kg/h. Assuming negligible nozzle lip, same air flow in both cases and constant coefficient of flow determine
   (*i*) The throat pressure with cleaner fitted ;
   (*ii*) The A / F ratio with cleaner fitted. **[Ans.** (*i*) 0.772 bar ; (*ii*) 13.67]

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Disadvantages :

This method is *not-used* now-a-days due to the following *reasons/disadvantages* :

(*i*) It requires a high pressure multi-stage compression. The large number of parts, the intercooler etc. make the system complicated and expensive.

(*ii*) A separate mechanical linkage is required to time the operation of fuel valve.

(*iii*) Due to the compression and the linkage the bulk of the engine increases. This also results in reduced B.P. due to power loss in operating the compression and linkage.

(*iv*) The fuel in the combustion chamber burns very near to injection nozzle which many times leads to overheating and burning of valve and its seat.

(*v*) The fuel valve sealing requires considerable skill.

(*vi*) In case of sticking of fuel valve the system becomes quite dangerous due to the presence of high pressure air.

12.4.2. Solid or Airless Injection

Injection of fuel directly into the combustion chamber without primary automisation is termed as **solid injection.** It is also termed as *mechanical injection.*

Main Components :

The *main components* of a fuel injection system are :

(*i*) *Fuel tank* ;

(*ii*) *Fuel feed pump* to supply the fuel from the main fuel tank to the injection pump ;

(*iii*) *Fuel filters* to prevent dust and abrasive particles from entering the pump and injectors ;

(*iv*) *Injection pump* to meter and pressurise the fuel for injection ;

(*v*) *Governor* to ensure that the amount of fuel is in accordance with variation in load ; and

(*vi*) *Fuel pipings and injectors* to take the fuel from the pump and distribute it in the combustion chamber by atomising it in fine droplets.

Main types of modern fuel injection systems :

1. Common-rail injection system.
2. Individual pump injection system.
3. Distributor system.

Atomisation of fuel oil has been secured by (*i*) *air blast* and (*ii*) *pressure spray.* Early diesel engines used air fuel injection at about 70 bar. This is sufficient not only to inject the oil, but also to atomise it for a rapid and thorough combustion. The expense of providing an air compressor and tank lead to the development of "*solid*" injection, using a liquid pressure of between 100 and 200 bar which is sufficiently high to atomise the oil it forces through spray nozzles. Great advances have been made in the field of solid injection of the fuel through research and progress in fuel pump, spray nozzles, and combustion chamber design.

1. Common-rail injection system :

Two types of common-rail injection systems are shown in Fig. 12.1 and 12.2 respectively.

- Refer Fig. 12.1. A single pump supplies high-pressure fuel to header, a relief valve holds pressure constant. The control wedge adjusts the lift of mechanical operated valve to set amount and time of injection.
- Refer Fig. 12.2. Controlled-pressure system has pump which maintains set head pressure. Pressure relief and timing valves regulate injection time and amount. Spring loaded spray valve acts merely as a check.

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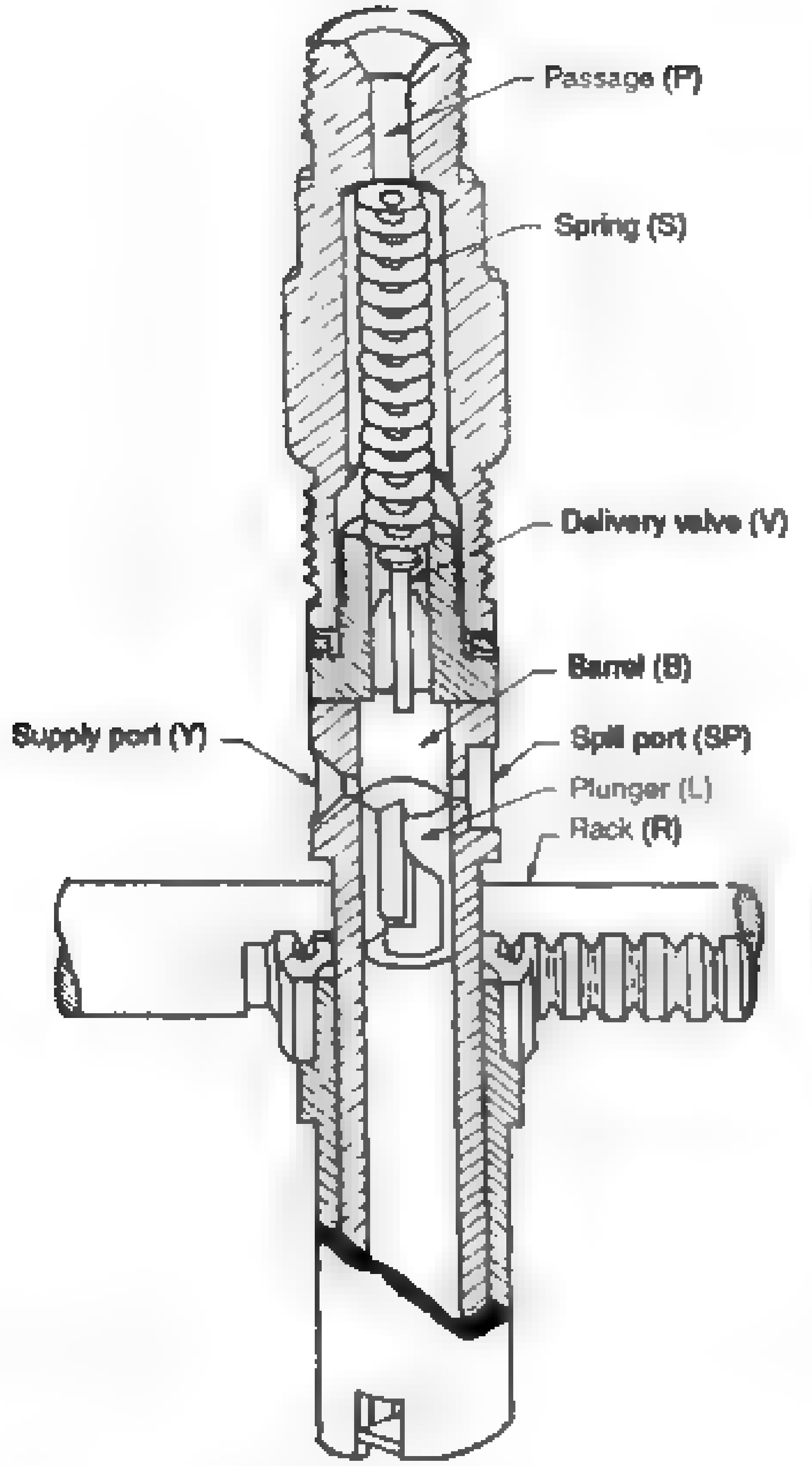

Fig. 12.5. Fuel pump.

- When the plunger is at its bottom stroke the ports SP and Y are uncovered (as shown in Fig. 12.5) oil from low pressure pump (not shown) after being filtered is forced into the barrel. When the plunger moves up due to cam and tappet mechanism, a stage reaches when both the ports SP and Y are closed and with the further upward movement of the plunger the fuel gets compressed. The high pressure thus developed lifts the delivery valve off its seats and fuel flows to atomiser through the passage P. With further rise of the plunger, at a certain moment, the port SP is connected to the fuel in the upper part of the plunger through the rectangular vertical groove by the helical groove, as a result of which a sudden drop in pressure occurs and the delivery valve falls back and occupies its seat against the spring force. The plunger is rotated by the rack R which is moved in or out by the governor. By changing the angular position of the helical groove (by rotating

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$$1.963 \times 10^{-5} = \left[\frac{\pi}{4} d_0^2 \times 6\right] \times 151.23 \times \left[\frac{30}{360} \times \frac{60}{1500}\right] \times \frac{750}{60} = 29.694\ d_0^2$$

$$\therefore \quad d_0 = \left(\frac{1.963 \times 10^{-5}}{29.694}\right)^{1/2} = 8.13 \times 10^{-4}\,\text{m} \quad \text{or} \quad \mathbf{0.813\ mm.} \quad \textbf{(Ans.)}$$

Example 12.3. *Fuel injection in a single cylinder, 4-stroke cycle C.I. engine running at 650 r.p.m. takes place through a single orifice nozzle and occupies 28° of crank travel. The fuel consumption of the engine is 2.2 kg/hour and the fuel used has a specific gravity of 0.875. If injection pressure is 150 bar and the combustion chamber pressure is 32 bar estimate the volume of fuel injected per cycle and the diameter of the orifice. Take coefficient of discharge of orifice = 0.88.*

Solution. *Given :* $n = 1$, $N = 650$ r.p.m. ; $\theta = 28°$ of crank travel ;

Fuel consumption = 2.2 kg/h ; Sp. gr. = 0.875 ;

$$\Delta p = p_1 - p_2 = 150 - 32 = 118 \text{ bar} ;\ C_d = 0.88$$

Volume of fuel injected per cycle :

$$\text{Fuel to be injected per cycle} = \frac{\text{Fuel consumption per cylinder}}{\text{No. of cycles per min.}}$$

$$= \frac{(2.2/60)}{(650/2)} = 1.128 \times 10^{-4}\ \text{kg}$$

$$\textit{Volume of fuel injected per cycle} = \frac{\text{Mass of fuel injected per cycle}}{\text{Density of fuel } (\rho_f)}$$

$$= \frac{1.128 \times 10^{-4}}{0.875 \times 1000} = 1.289 \times 10^{-7}$$

$$= \mathbf{0.1289\ cm^3.} \quad \textbf{(Ans.)}$$

Diameter of the orifice, d_0 :

$$\text{Time for fuel injection per cycle} = \left(\frac{\theta}{360} \times \frac{60}{N}\right) \text{sec.}$$

$$= \frac{28}{360} \times \frac{60}{650} = 0.00718\,\text{s}$$

Mass of fuel injected per second,

$$m_f = \frac{\text{Fuel injected per cycle}}{\text{Time for fuel injection}} = \frac{1.128 \times 10^{-4}}{0.00718} = 0.0157\ \text{kg/s}$$

Actual velocity of per cycle injection,

$$V_f = C_f \sqrt{\frac{2\Delta p}{\rho_f}} = 0.88 \sqrt{\frac{2 \times 118 \times 10^5}{(0.875 \times 1000)}} = 144.5\ \text{m/s}$$

Now,

$$m_f = A_0 \times V_f \times \rho_f$$

or

$$0.0157 = \left(\frac{\pi}{4} d_0^2\right) \times 144.5 \times (0.875 \times 1000)$$

or

$$d_0 = \left[\frac{0.0157 \times 4}{\pi \times 144.5 \times (0.875 \times 1000)}\right]^{1/2} = 3.976 \times 10^{-4}\ \text{m} = \mathbf{0.4\ mm.} \quad \textbf{(Ans.)}$$

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$$= \frac{\pi}{4} d_0^2 \times C_f \sqrt{2\Delta p \times \rho_f}$$

$$= \frac{\pi}{4} d_0^2 \times 0.67 \sqrt{2 \times 110 \times 10^5 \times 960} = 7238111 d_0^2$$

or $d_0 = 5.637 \times 10^{-4}$ m or **0.5637 mm. (Ans.)**

Example 12.8. *In a diesel fuel injection pump, the volume of fuel in the pump barrel before commencement of the effective stroke is 7 c.c. The diameter of the fuel line from pump to injector is 3 mm and is 700 mm long. The fuel in the injection valve is 2 c.c.*

(i) To deliver 0.10 c.c. of fuel at a pressure of 150 bar, how much displacement the plunger undergoes ? Assume a pump inlet pressure of 1 bar ;

(ii) What is the effective stroke of the plunger if its diameter is 7 mm.

Assume coefficient of compressibility of oil as 78.8 × 10⁻⁶ per bar at atmospheric pressure.

Solution. *Given* : The volume of fuel in the pump barrel before commencement of the effective stroke = 7 c.c.

The diameter and length of the fuel line from pump to injector = 3 mm, 700 mm,

Volume of fuel in the injection valve = 2 c.c.

Volume of fuel to be delivered = 0.10 c.c.

The pressure at which fuel to be delivered, p_1 = 150 bar

Atmospheric pressure, $p_2 = 1$ bar

Coefficient of compressibility, $C_c = 78.8 \times 10^{-6}$ per bar at atmospheric pressure

Diameter of plunger, $d_p = 7$ mm

(i) Displacement of plunger :

Coefficient of compressibility of oil,

$$C_c = \frac{\text{Change in volume per unit volume}}{\text{Difference in pressure causing compression}}$$

$$= \frac{(V_1 - V_2)}{V_1(p_1 - p_2)}$$

Total initial fuel volume,

V_1 = Volume of fuel in barrel + volume of fuel in the delivery line + volume of fuel in the injection valve

$$= 7 + \frac{\pi}{4}(0.3)^2 \times 70 + 2 = 13.95 \text{ c.c.}$$

No pressure is built up till the pump plunger closes the inlet port. Further advance of plunger will compress the fuel oil and raise the pressure to a required value. Once the delivery pressure is attained, further movement of plunger results in delivery of fuel oil at constant pressure.

Change in volume due to compression $= C_c(p_1 - p_2) \times V_1$

or $(V_1 - V_2) = 78.8 \times 10^{-6} \times (150 - 1) \times 13.95$

$= 0.16379$ c.c.

Total displacement of plunger

$= (V_1 - V_2) + 0.1 = 0.16379 + 0.1 =$ **0.26379 c.c. (Ans.)**

(ii) Effective stroke of the plunger, l_p :

$$\frac{\pi}{4} d_p^2 \times l_p = 0.26379 \quad \text{or} \quad \frac{\pi}{4} \times (0.7)^2 \times l_p = 0.26379$$

$\therefore$ $l_p = \dfrac{0.26379 \times 4}{\pi \times (0.7)^2} = 0.6854$ cm or **6.854 mm. (Ans.)**

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13

Ignition Systems (S.I. Engines)

13.1. Introduction. 13.2. Requirements of an ignition system. 13.3. Basic ignition systems. 13.4. Battery (or coil) ignition system. 13.5. Magneto ignition system. 13.6. Firing order. 13.7. Ignition timing. 13.8. Spark plugs. 13.9. Limitations of conventional ignition. 13.10. Electronic ignition systems—Highlights—Objective Type Questions—Theoretical Questions.

13.1. INTRODUCTION

- In S.I. engines the combustion process is initiated by a spark between the two electrodes of spark plug. This occurs just before the end of compression stroke. The ignition process must add necessary energy for *starting* and *sustaining* burning of the fuel till combustion takes place.
- Ignition is only *a pre-requisite of combustion. It does not influence the gross combustion process.* It is only a small scale phenomenon taking place within a specified small zone in the combustion chamber.
- Ignition only ensures initiation of combustion process and has no degree intensively or extensively.

Energy requirements for ignition :

— A spark energy below 10 millijoules is adequate to initiate combustion for A / F ratio 12-13 : 1 (Range of mixtures normally used) ; the duration of few micro-seconds is sufficient to start combustion.

— A spark can be struck between the gap in the two electrodes of the spark plug by sufficiently high voltage. There is a critical voltage called *breakdown voltage* below which no sparking would occur. In practice the pressure, temperature and density have a profound influence on the voltage required to cause the spark. Also, the striking voltage is increased due to the fouling factor of the electrodes owing to deposits and abrasion.

— For automotive engines, in normal practice, the spark energy to the tune of 40 millijoules and duration of about 0.5 millisecond is sufficient over entire range of operation.

13.2. REQUIREMENTS OF AN IGNITION SYSTEM

For an ignition system to be acceptable it must be *moderately priced, reliable and its performance must be adequate to meet all the demands imposed on it by various operating conditions.*

An ignition system should *fulfill the following requirements* :

1. It should have an adequate reserve of secondary voltage and ignition energy over the entire operating speed range of the engine.

2. It should consume the minimum of power and convert it efficiently to a high-energy spark across the spark-plug electrode gap.

3. It should have a spark duration which is sufficient to establish burning of the air-fuel mixture under all operating conditions.

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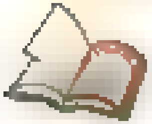

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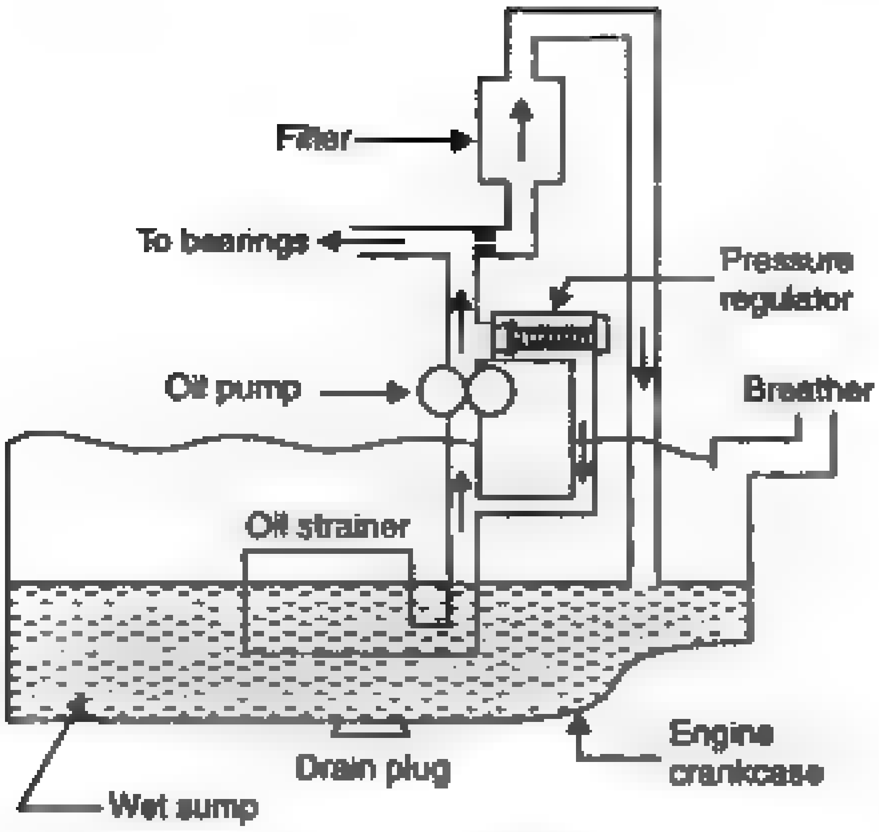

Fig. 14.9. Wet sump lubrication system.

14.6.3. Dry Sump Lubrication System

- Refer Fig. 14.10. In this system, the oil from the sump is carried to a separate storage tank outside the engine cylinder block. The oil from sump is pumped by means of a sump pump through filters to the storage tank. Oil from storage tank is pumped *to the engine cylinder* through oil cooler. Oil pressure may vary from 3 to 8 bar.
- *Dry sump lubrication system is generally adopted for high capacity engines.*

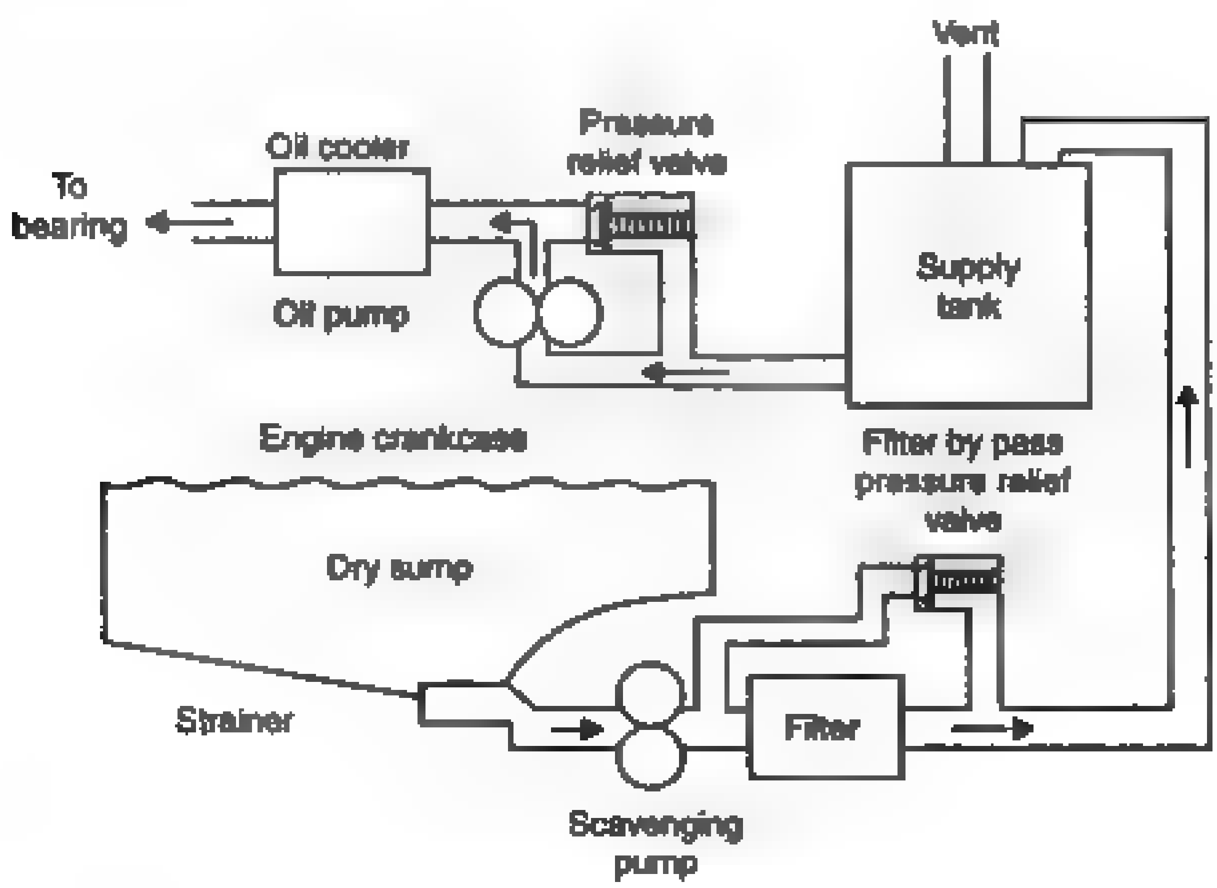

Fig. 14.10. Dry sump lubrication system.

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14.7. CRANKCASE VENTILATION

Crankcase ventilation is required owing to the following *two reasons* .

(i) The various contaminants such as water, gasoline, blowby gases etc. enter the crankcase due to several reasons, and may cause sludge and corrode metal parts

(ii) To relieve any pressure build-up in the crankcase which may cause leakage of the crankshaft seal.

In practice, following two types of ventilation systems are used :

1. Open system 2. Closed system.

1. Open system :

- In this system, fresh air supply is inducted into the crankcase during the compression stroke (due to creation of small vacuum). The entering air picks up the contaminants (water vapour, gases and H_2SO_4 vapour) and discharge them to the atmosphere during expansion stroke.
- The main *disadvantage* of this system is that the *natural ventilation is quite inadequate during idling or running at low speeds.*

2 Closed system :

- In closed system the *fresh air supply is taken to the crankcase from the carburettor.*
- Air cleaner and the breather outlets are connected to the intake manifold through a PCV valve to ensure the burning of all the crankcase gases in combustion chamber

HIGHLIGHTS

1. The difference between I.P. and B.P. is known as *total engine friction loss*
2. *Lubrication* is the admittance of oil between two surfaces having relative motion.
3. *Film lubrication* is that type of lubrication in which bearing surfaces are completely separated by a layer of film of lubricant and that the frictional resistance arises only due to relative movement of the lubricant layers.
4. If the oil film becomes thin enough so as not to support the load without occassional metal-to-metal contact then journal friction developed is called *boundary friction* and the lubrication existing in this range is known as *boundary lubrication*
5. *Viscosity* is the ability of the oil to resist internal deformation due to mechanical stresses and hence it is a measure of the ability of the oil film to carry a load.
6. *The mineral oils are very commonly used for all lubricating purposes*
7. *Lubricating grease* is a solid to semi-solid dispersion of a thickening agent in liquid lubricant.
8. Lubrication systems are classified as follows :
   (i) Wet sump lubrication system
   (ii) Dry sump lubrication system
   (iii) Mist lubrication system used for *two stroke engines*

OBJECTIVE TYPE QUESTIONS

Fill in the blanks or Say "Yes" or "No" :

1. Engine friction is defined as the difference between I.P. and ....
2. The pumping loss in two-stroke cycle engines is quite significant.
3. .... losses are caused due to the leakage of combustion products past the piston from the cylinder into the crankcase.

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15

Engine Cooling

15.1. Necessity of engine cooling. 15.2. Areas of heat flow in engines. 15.3. Gas temperature variation. 15.4. Heat transfer, temperature distribution and temperature profiles—Heat transfer—Temperature distribution—Temperature profiles. 15.5. Effects of operating variables on engine heat transfer. 15.6. Cooling air and water requirements. 15.7. Cooling systems—Air cooling system—Water-liquid cooling system 15.8. Components of water cooling system—Highlights–Objective Type Questions—Theoretical Questions.

15.1. NECESSITY OF ENGINE COOLING

In an I.C. engine, the temperature of the gases inside the engine cylinder may vary from 35°C or less to as high as 2750°C during the cycle. If an engine is allowed to run without external cooling, the cylinder walls, cylinder and pistons will tend to assume the average temperature of the gases to which they are exposed, which may be of the order of 1000 to 1500°C. Obviously at such high temperature ; the *metals will loose their characteristics and piston will expand considerably and sieze the liner. Of course theoretically thermal efficiency of the engine will improve without cooling but actually the engine will sieze to run.* If the cylinder wall temperature is allowed to rise above a certain limit, about 65°C, the lubricating oil will begin to evaporate rapidly and both cylinder and piston may be damaged. Also high temperature may cause excessive stress in some parts rendering them useless for further operation. In view of this, part of the heat generated inside the engine cylinder is allowed to be carried away by the cooling system. *Thus cooling system is provided on an engine for the following* **reasons :**

1. The even expansion of piston in the cylinder may result in seizure of the piston.
2. High temperatures reduce strength of piston and cylinder liner.
3. Overheated cylinder may lead to preignition of the charge, in case of spark ignition engines.
4. Physical and chemical changes may occur in lubricating oil which may cause sticking of piston rings and excessive wear of cylinder.
5. If the cylinder head temperature is high the volumetric efficiency and hence the power output of the engine is reduced.

Thus engine cooling is required to keep the temperature of the engine low in order to *avoid* :

(*i*) Loss of volumetric efficiency and hence power ;

(*ii*) Engine seizure ;

(*iii*) Danger of engine failure.

- *Almost 25 to 35 percent of total heat supplied in the fuel is removed by the* **cooling medium.**
- Heat carried away by *lubricating oil and heat lost by radiation* amounts to *3 to 5 per cent of the total heat supplied.*

  It must be noted that *heat carried away by the coolant is a dead loss because not only no useful work can be obtained from it but a part of the engine power is also used to remove this heat.* Hence, it is of paramount importance that this loss is *kept minimum* by the designer.

$$Re = \text{Reynold's number} \left(\frac{\rho V D}{\mu}\right),$$

$$Pr = \text{Prandtl number} \left(\frac{c_p \mu}{k}\right),$$

$\frac{D}{L}$ = Diameter to length ratio,

C = A constant to be determined experimentally,

c_p = Specific heat at constant pressure,

k = Thermal conductivity,

ρ = Density,

μ = Dynamic viscosity, and

V = Velocity.

The overall heat transfer coefficient :

While dealing with the problems of fluid to fluid heat transfer across a metal boundary, it is usual to adopt an overall heat transfer coefficient U which *gives the heat transmitted per unit area-unit time per degree temperature difference between the bulk fluids on each side of the metal.*

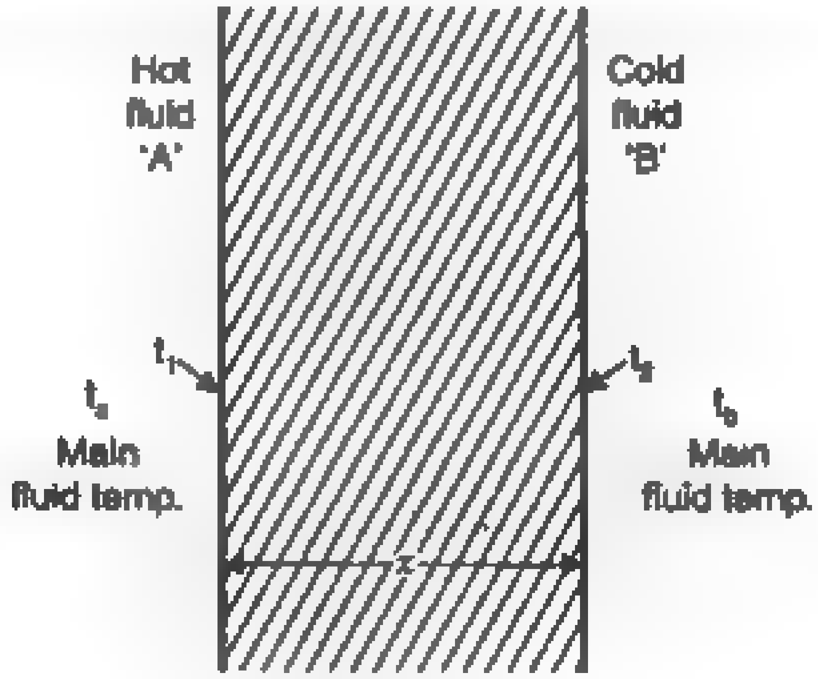

Fig. 15.2

Refer Fig. 15.2.

Let, h_a = Heat transfer coefficient from hot fluid to metal surface,

h_b = Heat transfer coefficient from metal surface to cold fluid, and

k = Thermal conductivity of metal wall.

The equations of heat flow through the fluids and the metal surface are as follows :

$$Q = h_a A (t_a - t_1) \quad ...(i)$$

$$Q = \frac{kA (t_1 - t_2)}{x} \quad ...(ii)$$

$$Q = h_b A (t_2 - t_b) \quad ...(iii)$$

By rearranging the equations (*i*), (*ii*) and (*iii*), we get

$$t_a - t_1 = \frac{Q}{h_a A} \quad ...(iv)$$

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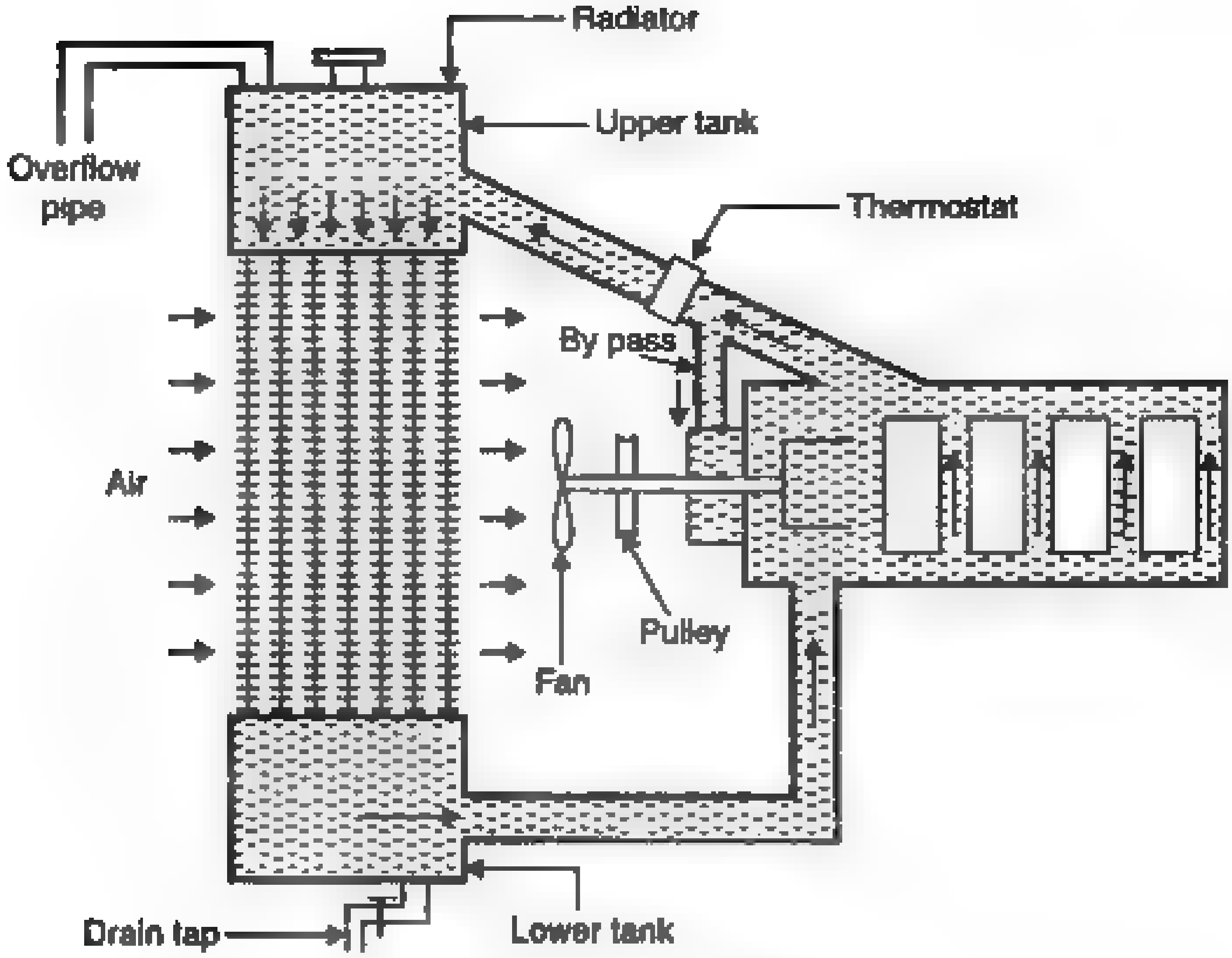

Fig. 15.17. Thermostatically controlled cooling system.

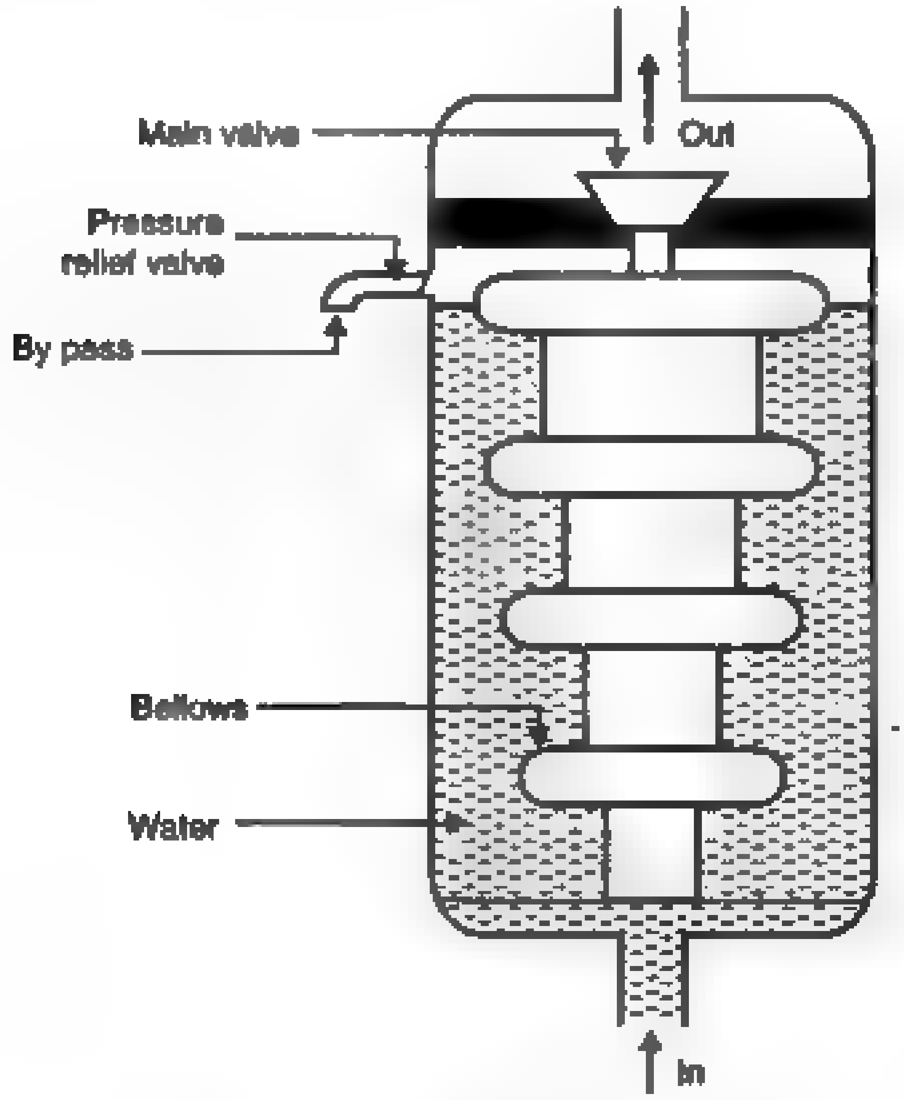

Fig. 15.18. Typical thermostat.

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$\therefore$ $$\text{Heat supplied} = \frac{\text{Work developed}}{\eta_{petrol}} = \frac{90}{0.25} = 360 \text{ kW or kJ/s}$$

Energy/heat going to cooling water = 360 × 0.32 = 115.2 kJ/s

Also, $$115.2 = m_w \times c_{pw} \times \Delta t_w$$

(where, m_w = mass of cooling water required ; c_{pw} = specific heat of water at constant pressure)

$\therefore$ $$m_w = \frac{115.2}{c_{pw} \times \Delta t_w} = \frac{115.2}{4.187 \times 27} = 1.019 \text{ kg/s or } \mathbf{3668\ kg/h.} \quad \textbf{(Ans.)}$$

Diesel engine :

Heat supplied $\dfrac{90}{0.3}$ = 300 kW or kJ/s

Heat going to cooling water = 300 × 0.28 = 84 kJ/s

Also, $84 = m_w \times 4.87 \times 27$

$\therefore$ $$m_w = \frac{[illegible]}{4.187 \times 27} = 0.743 \text{ kg/s or } \mathbf{2674.9\ kg/h.} \quad \textbf{(Ans.)}$$

HIGHLIGHTS

1. Almost 25 to 35% of total heat supplied in the fuel is removed by the cooling medium.
2. At least 95% of the total heat transfer between the working fluid and engine components and the engine components and cooling fluid is effected by "forced convection" process of heat transfer.
3. When the spark advance is different from the optimum value the heat rejected to cooling system is increased.
4. In *air-cooling* system, heat is carried away by the air flowing over and around the cylinder. Here fins are cast on the cylinder head and cylinder barrel which provide additional conductive and radiating surface.
5. In *water-cooling* system of cooling engines, the cylinder walls and heads are provided with jacket through which the cooling liquid can circulate.
6. The *methods* used for circulating water around the cylinder and cylinder head :
   (*i*) Thermo-system cooling
   (*ii*) Forced or pump cooling
   (*iii*) Cooling with thermostatic regulator
   (*iv*) Pressurised water cooling
   (*v*) Evaporative cooling.

OBJECTIVE TYPE QUESTIONS

Fill in the blanks or Say "Yes" or "No".

1. If the cylinder head temperature is high the volumetric efficiency and hence the power output of the engine is ....
2. Almost 25 to 35 percent of total heat supplied in the fuel is removed by the cooling medium.
3. Heat carried away by lubricating oil and heat lost by radiation amounts to 10 to 15 percent of the total heat supplied.
4. Heat carried away by the coolant is a dead loss.
5. At very low temperature, starting of engine becomes difficult.
6. Undercooling shortens valve life.
7. The overall heat transfer coefficient gives the heat transmitted per unit area-unit time per degree temperature difference between the bulk fluids on each side of the metal .

(ii) *The temperature of the charge is raised as it is compressed, resulting in a higher temperature within the cylinders.* This is partially beneficial in that it helps to produce better vapourisation of fuel (in case of S.I. engines) but deterimental in that it tends to lessen the density of the charge. The increase in temperature of the charge also effects the detonation of the fuel.

Supercharging tends to increase the possibility of detonation in a S.I. engine and lessen the possibility in a C.I. engine.

(iii) Power is required to drive the supercharger. This is usually taken from the engine and thereby removes, from over-all engine output, some of the gain in power obtained through supercharging.

Compressors used are of the following three types :

(i) ***Positive displacement type*** used with many reciprocating engines in stationary plants, vehicles and marine installations,

(ii) ***Axial flow type*** seldom used to supercharge reciprocating engines, it is widely used as the compressor unit of the *gas turbines.*

(iii) ***Centrifugal type*** *widely used* as the *supercharger for reciprocating engines, as well as compressor for gas turbines.* It is almost exclusively used as the supercharger with reciprocating power plants for *aircraft because it is relatively light and compact, and produces continuous flow rather than pulsating flow as in some positive displacement types.*

- A *correctly matched supercharger* will raise the cylinder's brake mean effective pressure (b.m.e p.) *to well above that of a naturally aspirated engine without creating excessively high peak cylinder pressures ; the actual increase in the brake mean effective pressure is basically determined by the level of boost pressure the supercharged system is designed to deliver.*
- *Large commercial vehicle diesel engines are frequently turbocharged, with the objectives of raising b.m.e.p. (and therefore torque and power output) and at the same time reducing the engine's maximum speed. The other benefits of raising the cylinder mean pressure and decreasing the engine's limiting speed is that the engine mechanical losses and noise are reduced and there is an improvement in fuel consumption, normally, an added bonus of prolonged engine life expectancy.*

Object of supercharging. The objects of supercharging include one or more of the following :

1. *To increase the power output for a given weight and bulk of the engine* .... relates to *aircraft, marine and automotive engines.*
2. *To compensate for the loss of power due to altitute* .... Relates to *aircraft and other engines which are used at high altitudes.*
3. *To obtain more power from an existing engine.*

Effects of supercharging on performance of the engine :

1. The '*power output*' of a supercharged engine is *higher* than its naturally aspiraced counterpart.
2. The '*mechanical efficiencies*' of supercharged engines are *slightly better* than the naturally aspirated engines.
3. Inspite of better mixing and combustion due to reduced delay a mechanically supercharged otto engine almost always have '*specific fuel consumption*' *higher than a naturally aspirated engine.*

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16.2.6. Thermodynamic Cycle and Supercharging Power

Fig. 16.6 shows the thermodynamic cycle of a supercharged I.C. engine on the *p-v* diagram for an *ideal otto cycle*.

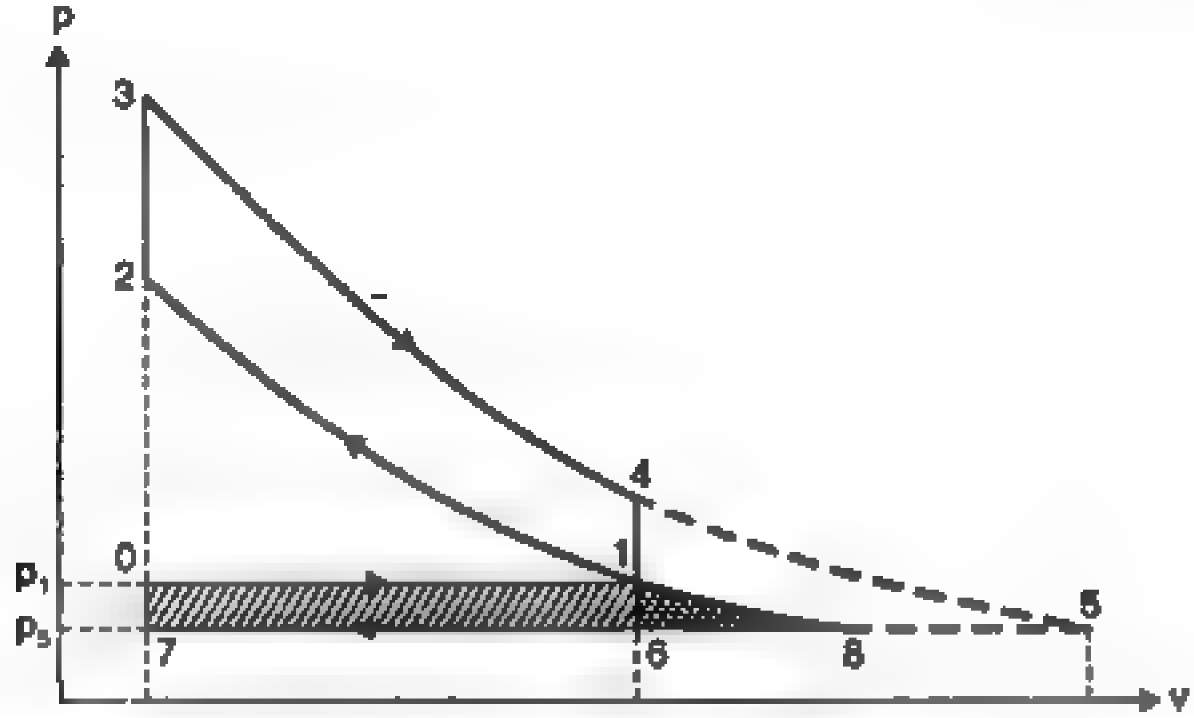

Fig. 16.6. Thermodynamic cycle of supercharged engine on *p-v* diagram for an ideal otto cycle.

- The pressure p_1 represents the *supercharging pressure* and p_5 is the *exhaust pressure.*
- The thermodynamic cycle, consists of the following *processes :*
  - (*i*) **0–1.** Admission of air *at the supercharging pressure* (which is *greater than* atmospheric pressure).
  - (*ii*) **1–2.** Isentropic compression.
  - (*iii*) **2–3.** Heat addition at constant volume (for diesel cycle, this will be replaced by a constant pressure process, representing heat addition at constant pressure.
  - (*iv*) **3–4.** Isentropic expansion.

    4–1–6. Heat rejection at constant volume (blow down to atmospheric pressure).

    6–7. Driving out exhaust at constant atmospheric pressure.

The thermodynamic cycle for the *supercharger* consists of the following *processes* :

(*i*) **7–6–8.** Admission of air at atmospheric pressure.

(*ii*) **8–1.** Isentropic compression to pressure p_1.

(*iii*) **1–0.** Delivery of supercharged air, at a constant pressure p_1.

- Area 8-6-7-0-1-8 represents the *supercharger work* (mechanically driven) in supplying air at a pressure p_1, while the area 1-2-3-4-1, is the *output of the engine*. Area 0-1-6-7-0 represents the *gain in work during the gas exchange process due to supercharging. Thus a part of the supercharger work is recovered.* However, the work represented by the area 1-6-8-1 *cannot be recovered and represents a loss of work.*

Supercharging power :

Refer Fig. 16.7.

p_1, v_1, T_1 = Initial conditions of air at entry to the supercharger ;

p_2, v_2, T_2 = Final conditions of air at exit from the supercharger.

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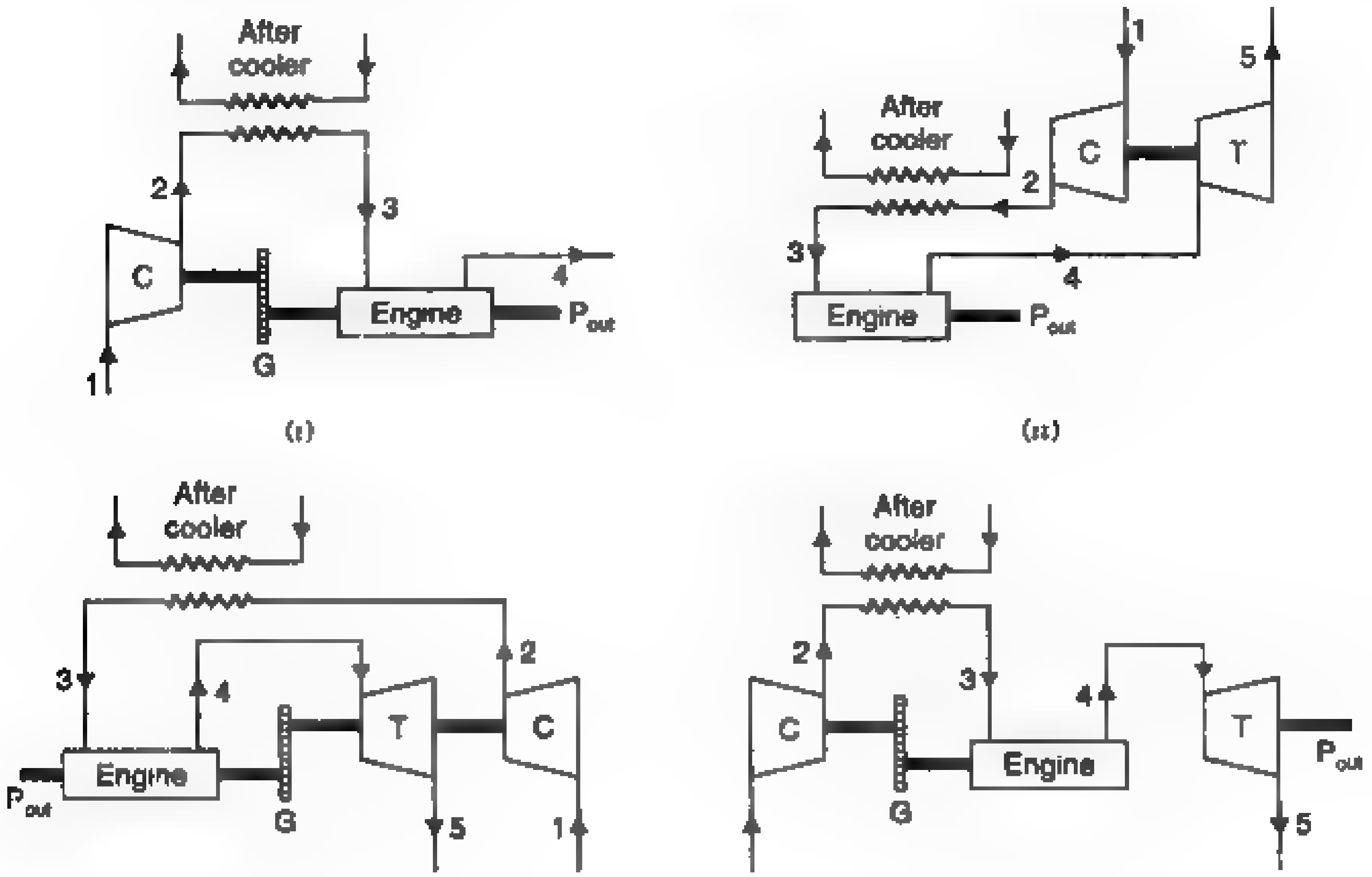
particularly at part loads, then the remaining load of the compressor is taken care of by the engine



(i) (ii) (iii) (iv)

C = Compressor
T = Turbine
G = Gearing

1. Inlet to compressor
2. Outlet from compressor
3. Inlet to engine
4. Exhaust from engine (and inlet to turbine)
5. Exhaust from turbine

Fig. 16.9. Supercharging arrangements.

- Fig. 16.9 (*iv*) shows an arrangement of supercharging in which engine supplies its total power to the compressor and the exhaust gases from the engine run the turbine giving the power output. Such plants are called *fuel-piston engines*.

16.7. TURBOCHARGERS

16.7.1. Introduction

- ***Turbochargers** are centrifugal compressors driven by the exhaust gas turbines.* By utilising the exhaust energy of the engine it recovers a substantial part of energy which would otherwise go waste ; thus the *turbocharger will not draw upon the engine power*. These are *nowadays extensively used to supercharging almost all types of two stroke engines*.
- *A typical petrol engine may harness up to 30% of the energy contained in the fuel supplied to do useful work under optimum conditions but the remaining 70% of this energy is lost in the following way :*

$$= 0.9147 \times \frac{[redacted]}{1.044} = 1.2143 \text{ kg/s}$$

∴ Power required to run the supercharger,

$$P_{sup} = \dot{m}_{a(sup)} \times W_{sup}$$

$$= 1.2143 \times 74.48 = \mathbf{90.44\ kW.} \quad \textbf{(Ans.)}$$

Example 16.2. *A diesel engine operating on four-stroke cycle is to be designed to operate with following characteristics at sea level, where the mean conditions are 1.0132 bar and 10°C.*

B.P. = 260 kW, volumetric efficiency = 78% (at sea level free air conditions), specific fuel consumption = 0.247 kg/B.P.h. ; A/F ratio = 17 ; speed = 1500 r.p.m.

Calculate the required engine capacity and the anticipated brake mean effective pressure.

The engine is fitted with a supercharger so that it may be operated at an altitude of 2700 m where the atmospheric pressure is 0.72 bar. The power taken by a supercharger is 8 per cent of the total power produced by the engine and the temperature of the air leaving the supercharger is 32°C. The air-fuel ratio and thermal efficiency remain the same for the supercharged engine as when running unsupercharged at sea level, as does the volumetric efficiency. Calculate the increase of air pressure required at the supercharger to maintain the same net output of 260 kW. Take R = 0.287 kJ/kgK.

Solution. *Given :* $p_1 = 1.0132$ bar, $T_1 = 10 + 273 = 283$ K, B.P. = 260 kW,

$\eta_{vol.} = 78\%$, s.f.c. = 0.247 kg / B.P. h, A / F ratio = 1.7 : 1,

$N = 1500$ r.p.m.

Engine capacity :

Fuel consumption, $m_f = \dfrac{s.f.c. \times \text{B.P}}{60}$ kg/min. $= \dfrac{0.247 \times 260}{60} = 1.07$ kg/min.

Air consumption = Fuel consumption × A / F ratio

$= 1.07 \times 17 = 18.19$ kg / min.

Air consumption per stroke $= \dfrac{\text{Air consumption in kg / min.}}{\text{No. of cycles / min.}}$

$$= \frac{\text{Air consumption in kg/min.}}{N/2} = \frac{18.19}{1500/2} = 0.0242 \text{ kg}$$

Let V_s be the swept volume, then mass of free air corresponding to swept volume $= \dfrac{p_1 V_s}{RT_1}$

$$= \frac{(1.0132 \times 10^5) \times V_s}{287 \times 283} = 1.247\ V_s \text{ kg}$$

Volumetric efficiency, $\eta_{vol.} = \dfrac{\text{Mass of air taken in per stroke}}{\text{Mass of free air corresponding to swept volume}}$

$$0.78 = \frac{0.0242}{1.247\ V_s} \quad \text{or} \quad V_s = \frac{0.0242}{0.78 \times 1.247} = 0.02488 \text{ m}^3$$

i.e., *Engine capacity* $= \mathbf{0.02488\ m^3.}$ **(Ans.)**

Brake mean effective pressure, p_{mb} (bar) :

We know that B.P. $= \dfrac{p_{mb}\ LANk \times 10}{6}$ kW

$$260 = \frac{p_{mb} \times V_s \times Nk \times 10}{6} \text{ kW} \qquad (\because \ L \times A = V_s)$$

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L = Length of stroke, m,

A = Area of piston, m^2, and

$k = \frac{1}{2}$ for 4-stroke engine

= 1 for 2-stroke engine.

(ii) ***Brake power (B.P.).*** *The power developed by an engine at the output shaft is called the brake power.*

$$\text{B.P.} = \frac{2\pi NT}{60 \times 1000} \text{ kW} \qquad ...(17.2)$$

where, N = Speed in r.p.m., and

T = Torque in N-m.

The difference between I.P. and B.P. is called *frictional power, F.P.*

i.e.,

$$\text{F.P.} = \text{I.P.} - \text{B.P.} \qquad ...(17.3)$$

The ratio of B.P. to I.P. is called *mechanical efficiency*

i.e., *Mechanical efficiency,*

$$\eta_{mech.} = \frac{\text{B.P.}}{\text{I.P.}} \qquad ...(17.4)$$

2. Mean effective pressure and torque :

"Mean effective pressure" is defined as hypothetical pressure which is thought to be acting on the piston throughout the power stroke. If it is based on I.P. it is called *indicated mean effective pressure* ($I_{m.e.p.}$ or p_{mi}) and if based on B.P. it is called brake mean effective pressure ($B_{m.e.p.}$ or p_{mb}). Similarly, frictional mean effective pressure ($F_{m.e.p.}$ or p_{mf}) can be defined as :

$$F_{m.e.p.} = I_{m.e.p.} - B_{m.e.p.} \qquad ...(17.5)$$

The torque and mean effective pressure are related by the engine size.

Since the power (P) of an engine is dependent on its size and speed, therefore it is not possible to compare engine on the basis of either power or torque. *Mean effective pressure is the true indication of the relative performance of different engines.*

3. Specific output :

It is defined as the *brake output per unit of piston displacement* and is given by :

$$\text{Specific output} = \frac{\text{B.P.}}{A \times L}$$

$$= \text{Constant} \times p_{mb} \times \text{r.p.m.} \qquad ...(17.6)$$

For the same piston displacement and brake mean effective pressure (p_{mb}) an engine running at higher speed will give more output.

4. Volumetric efficiency :

It is defined as the ratio of actual volume (reduced to N.T.P.) of the charge drawn in during the suction stroke to the swept volume of the piston.

The average value of this efficiency is from 70 to 80 per cent but in case of *supercharged engine* it may be *more* than 100 per cent, if air at about atmospheric pressure is forced into the cylinder at a pressure greater than that of air surrounding the engine.

5. Fuel-air ratio :

It is the ratio of the mass of fuel to the mass of air in the fuel-air mixture.

Relative fuel air ratio is defined as the ratio of the actual fuel-air ratio to that of stoichiometric fuel-air ratio required to burn the fuel supplied.

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$$= 840\,AC_d \sqrt{\frac{h_w}{\rho_a}}\ \text{m}^3/\text{min}.$$

Mass of air passing through the orifice is given by

$$m_a = V_a \rho_a = 14 \times \frac{\pi d^2}{4 \times 100^2} \times C_d \sqrt{\frac{h_w}{\rho_a}} \times \rho_a$$

$$= 0.0011\ C_d \times d^2 \sqrt{h_w \rho_a}\ \text{kg/s}$$

$$= 0.066\ C_d \times d^2 \sqrt{h_w \rho_a}\ \text{kg/min.} \qquad ...(17.12)$$

(*ii*) ***Viscous-flow air meter :***

Alcock viscous-flow air meter is another design of air meter. It is not subjected to the errors of the simple types of flow meters. With the air-box the flow is proportional to the square root of the pressure difference across the orifice. With the Alcock meter the air flows through a form of honey-comb so that flow is viscous. *The resistance of the element is directly proportional to the air velocity and is measured by means of an inclined manometer.* Felt pads are fitted in the manometer connections to damp out fluctuations. The meter is shown in Fig. 17.2.

The accuracy is improved by fitting a damping vessel between the meter and the engine to reduce the effect of pulsations.

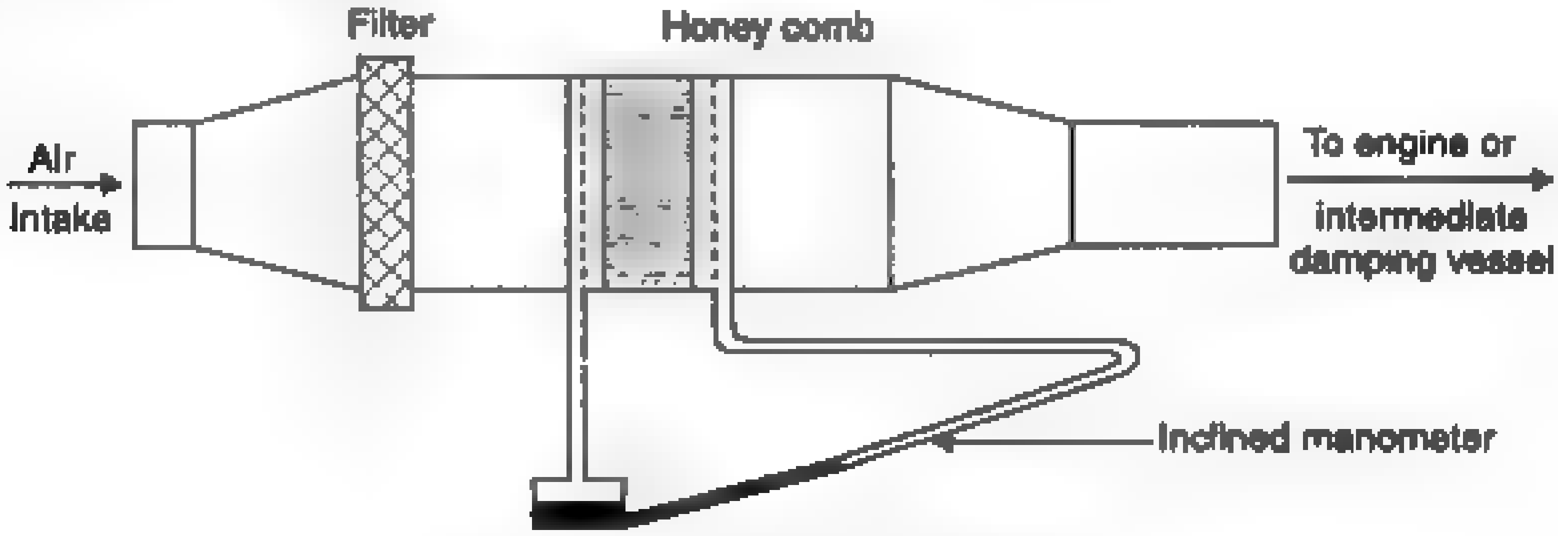

Fig. 17.2. Alcock viscous-flow air meter

4. **Measurement of exhaust smoke :**

The following smoke meters are used :

(*i*) Bosch smoke meter
(*ii*) Hatridge smoke meter
(*iii*) PHS smoke meter.

5. **Measurement of exhaust emission :**

Substances which are emitted to the atmosphere from any opening down stream of the exhaust part of the engine are termed as "exhaust emissions". Some of the more commonly used instruments for measuring exhaust components are given below :

(*i*) Flame ionisation detector
(*ii*) Spectroscopic analysers
(*iii*) Gas chromatography

6. **Measurement of B.P. :**

The B.P. of an engine can be determined by a brake of some kind applied to the brake pulley of the engine. The arrangement for determination of B.P. of the engine is known as *dynamometer*. The dynamometers are classified into following two classes .

(*i*) Absorption dynamometers
(*ii*) Transmission dynamometers.

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ignition setting or speed may by ascertained by producing a new family of curves. An alternative method of plotting these parameters is to use the air-fuel ratio as the abscissa. Here it can be seen that maximum economy occurs with a slightly weak mixture. This means that there is excess air and combustion is complete. Maximum power occurs with a slightly rich mixture when all the available oxygen is used. The I C. engine efficiency is the inverse of the specific fuel consumption with the constant calorific value as a factor. Thus the curves of specific fuel consumption (s.f.c.) also represent efficiency. The maximum value of brake I.C. engine efficiency for S.I. and C.I. engines are of the order 35% and 40% respectively.

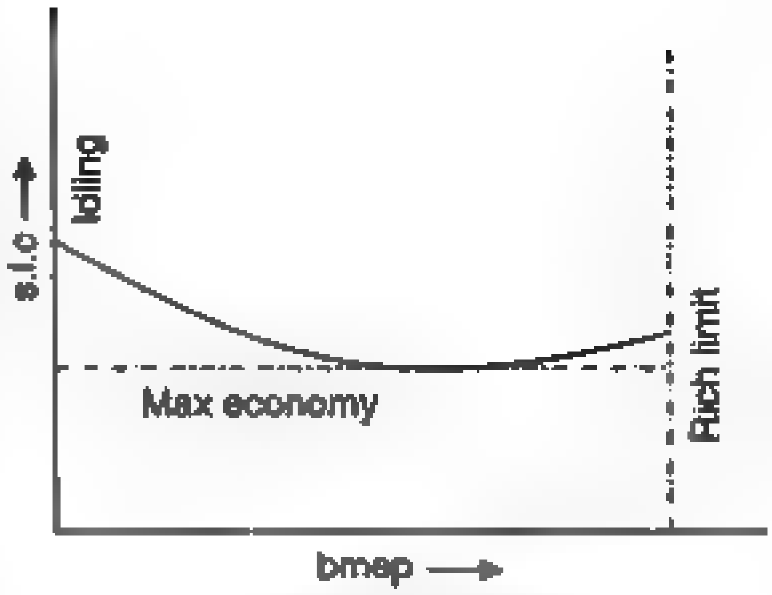

Fig. 17 7. Specific fuel consumption-brake mean effective pressure curve for the C.I. engine.

C.I. engine. The flat curve of Fig. 17.7 illustrates that at *part load the compression ignition engine is more economical than the spark ignition engine.* This is the benefit of quality control rather than quantity control of power.

17.5. COMPARISON OF PETROL AND DIESEL ENGINES—FUEL CONSUMPTION LOAD OUTPUTS AND EXHAUST COMPOSITION

I. Fuel Consumption :

Fig. 17.8 shows fuel consumption loops, for both petrol and diesel engines, plotted on a base of brake mean effective pressure (b.m.e.p).

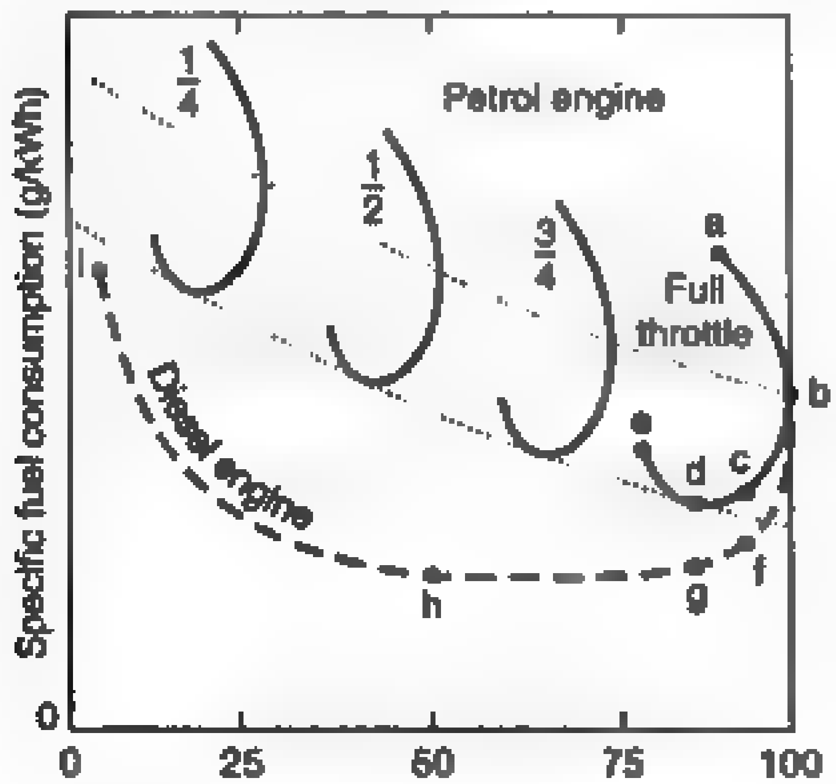

Fig. 17.8. Comparision of fuel consumption loops for petrol and diesel engines on a base of engine load (b.m.e.p.).

a = Excessively rich mixture gives slow and unstable combustion.

b = Maximum b.m.e.p. with something like 10–20% rich mixture.

c = Correct stoichiometric mixture of 14.7 : 1 by weight

d = Maximum thermal efficiency with something like 10–20% weak mixture (approaches ideal constant volume combustion).

e = Excessively weak mixture gives slow burning and popping back through air intake.

f = Maximum b.m.e.p. with satisfactory clear exhaust requires mixture strength of about 18 : 1 by weight.

g-h = Maximum thermal efficiency, minimum specific fuel consumption ranges between 50–85% of maximum b.m.e.p.

i = No-load (low speed idle) requires mixture strength 100–75 : 1 by weight.

- In case of a diesel *engine, load and speed output is controlled entirely by varying the quantity of fuel injected into the cylinder without misfiring occurring,* that is, from 0–100% of the maximum b.m.e.p. developed.
- With the petrol engines, however, if there was no throttle (full throttle position) the effects of varying the mixture strength from the richest position (a) to the weakest position

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D, L, displacement :

Power developed $= \dfrac{2\pi N T_b}{60 \times 1000}$ kW $= \dfrac{p_m LANk \times 10}{6}$ kW

i.e., $$\frac{2\pi N T_b}{60 \times 1000} = \frac{p_m LANk \times 10}{6}$$

Substituting the values, we get

$$\frac{2\pi \times 3000 \times 160}{60 \times 1000} = \frac{9.6 \times D \times \frac{\pi}{4} \times D^2 \times 3000 \times \frac{1}{2} \times 10}{6}$$

$$50.265 = 18849.6\, D^3$$

$$\therefore \quad D = \left(\frac{50.265}{18849.6}\right)^{1/3} = 0.1387 \text{ m or } \mathbf{138.7 \text{ mm.}} \quad \textbf{(Ans.)}$$

$$\therefore \quad \mathbf{L = D = 138.7 \text{ mm.}} \quad \textbf{(Ans.)}$$

Displacement $= \dfrac{\pi}{4} D^2 \times L = \dfrac{\pi}{4} \times 0.1387^2 \times 0.1387$

$= \mathbf{0.002096 \text{ m}^3.}$ **(Ans.)**

Example 17.5. *A turbocharged six-cylinder diesel engine has the following performance details :*

(i) *Work done during compression and expansion = 820 kW*
(ii) *Work done during intake and exhaust = 50 kW*
(iii) *Rubbing friction in the engine = 150 kW*
(iv) *Network done by turbine = 40 kW*

If the brake mean effective pressure is 0.6 MPa, determine the bore and stroke of the engine taking the ratio of bore to stroke as 1 and engine speed as 1000 r.p.m. **(GATE-1998)**

Solution. *Given :* $p_{mb} = 0.6$ MP$_a$ $= 6$ bar ; $\dfrac{D}{L} = 1$; $N = 1000$ r.p.m.

D, L :

Net work available $= 820 - (50 + 150 + 40) = 580$ kW

$$\text{B.P} = \frac{n \times p_{mb} LANk \times 10}{6}$$

$$580 = \frac{6 \times 6 \times D \times \frac{\pi}{4} D^2 \times 1000 \times \frac{1}{2} \times 10}{6} = 23562\, D^3$$

$$D^3 = \left(\frac{580}{23562}\right)^{1/3} = 0.2908 \text{ m} \quad \text{or} \quad 290.8 \text{ mm}$$

Hence D = L = 290.8 mm. (Ans.)

Example 17.6. *A spark-ignition engine, designed to run on octane (C_8H_{18}) fuel, is operated on methane (CH_4). Estimate the ratio of the power input of the engine with methane fuel to that with octane. In both cases the fuel ratio is stoichiometric, the mixture is supplied to the engine at the same conditions, the engine runs at the same speed, and has the same volumetric and thermal efficiencies. The heating value of methane is 50150 kJ/kg while that of octane in 44880 kJ/kg.* **(U.P.S.C.-1994)**

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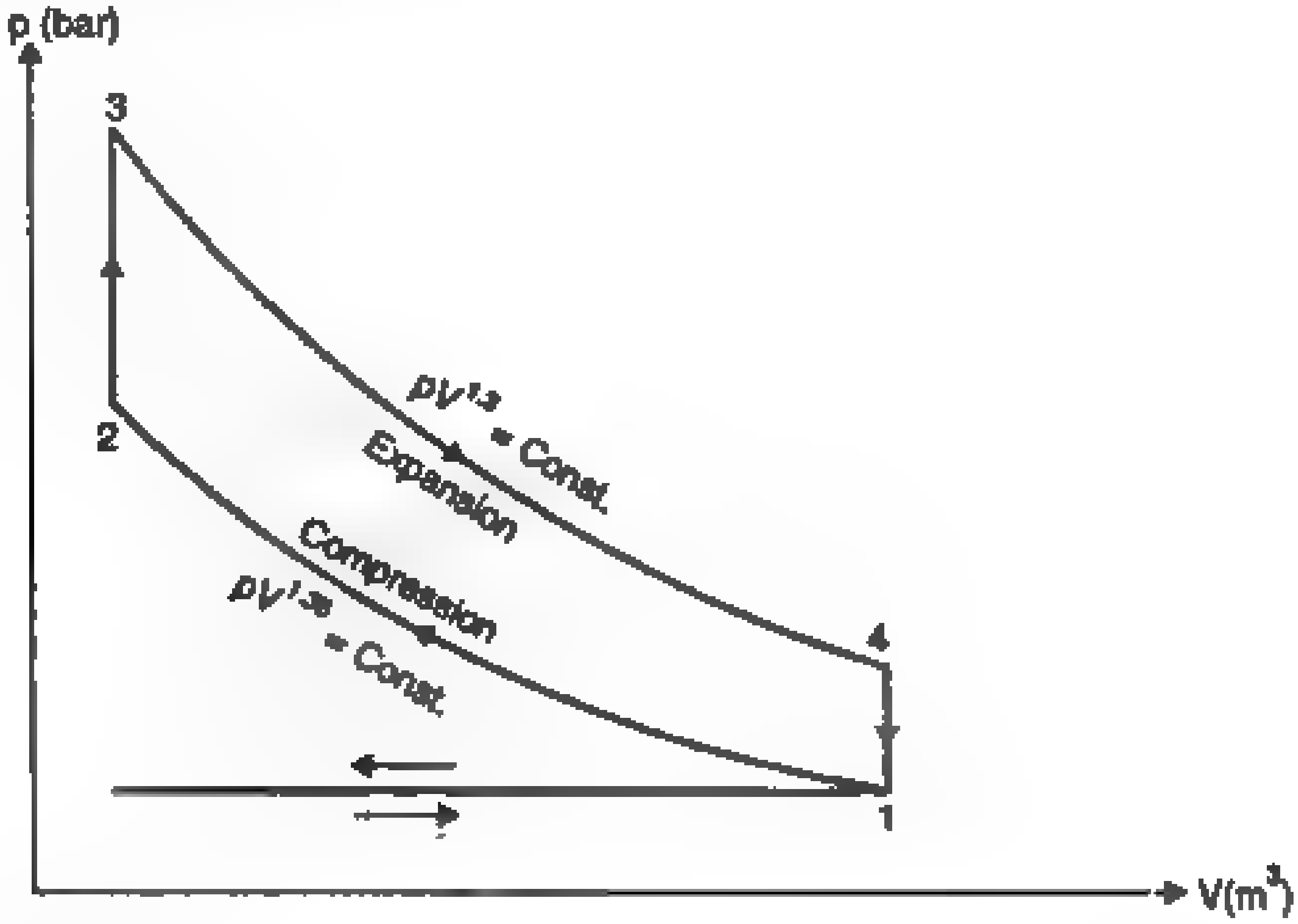

Fig. 17.12

$\therefore \qquad p_2 = p_1 \times 8.78 = 0.9 \times 8.78 = 7.9$ bar

To find p_4, considering *expansion process 3-4*, we have

$$p_3 V_3^{\ 1.3} = p_4 V_4^{\ 1.3}$$

or
$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^{1.3} = (5)^{1.3} = 8.1$$

$$\therefore \qquad p_4 = \frac{p_3}{8.1} = \frac{24}{8.1} = 2.96 \text{ bar}$$

Work done/cycle = Area 1-2-3-4

= Area under the curve 3-4 – area under the curve 1-2

$$= \frac{p_3V_3 - p_4V_4}{1.3 - 1} - \frac{p_2V_2 - p_1V_1}{1.35 - 1}$$

$$= \frac{p_3V_3 - p_4V_4}{0.3} - \frac{p_2V_3 - p_1V_4}{0.35} \qquad (\because \ V_1 = V_4 \text{ and } V_2 = V_3)$$

$$= \frac{10^5(24V_3 - 2.9V_4)}{0.3} - \frac{10^5(7.9V_3 - 0.9V_4)}{0.35}$$

$= 10^5\,[(80V_3 - 9.86V_4) - (22.57V_3 - 2.57V_4)]$

$= 10^5(80V_3 - 9.86V_4 - 22.57V_3 + 2.57V_4)$

$= 10^5(57.43V_3 - 7.29V_4)$

$$= 10^5(57.43V_3 - 7.29 \times 5V_3) \qquad \left[\because \ \frac{V_4}{V_3} = 5\right]$$

$= 10^5 \times 20.98\ V_3$ N-m.

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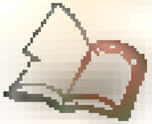

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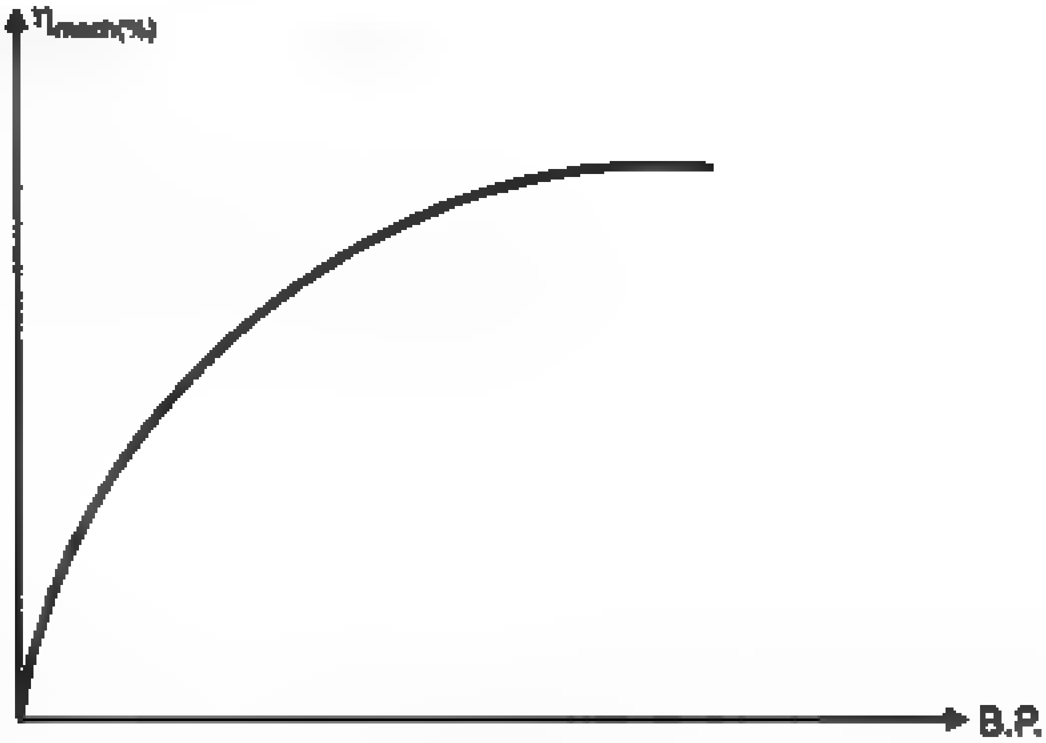

Fig. 17.16

Example 17.31. *During the trial of a gas engine following observations were recorded :*

Bore	*= 320 mm*
Stroke	*= 420 mm*
Speed	*= 200 r.p.m.*
Number of explosions/min.	*= 90*
Gas used	*= 11.68 m³/h*
Pressure of gas	*= 170 mm of water above atmospheric pressure*
Barometer	*= 755 mm (mercury)*
Mean effective pressure	*= 6.2 bar*
Calorific value of gas used	*= 21600 kJ/kg at N.T.P.*
Net load on brake	*= 2040 N*
Brake drum diameter	*= 1.2 m*
Ambient temperature	*= 25°C*

Calculate : (i) Mechanical efficiency, and (ii) Brake thermal efficiency.

Solution. $n = 1, D = 0.32$ m, $L = 0.42$ m, $N = 200$ r.p.m.,

$$Nk = 90,\ V_g = \frac{11.68}{3600} = 0.00324 \text{ m}^3/\text{s},$$

Pressure of gas $= 755 + \dfrac{170}{13.6} = 767.5$ mm Hg

$$p_{mi} = 6.2 \text{ bar},\ C = 21600 \text{ kJ/kg at N.T.P.}$$

$$(W - S) = 1840 \text{ N},\ D_b = 1 \text{ m}.$$

(i) Mechanical efficiency :

As the number of explosions per minute is given as 90 per minute and r.p.m. of engine is 200 it shows that the engine is operating on four-stroke cycle.

Indicated power (I.P.) is given by the relation :

$$\text{I.P.} = \frac{np_{mi}LANk \times 10}{6}$$

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Calorific value of fuel	= 45100 kJ/kg
Per cent carbon in the fuel	= 85%
Per cent hydrogen in the fuel	= 15%
Pressure of air at the end of suction stroke	= 1.013 bar
Temperature at the end of suction stroke	= 25°C

Calculate : (i) Brake mean effective pressure, (ii) Specific fuel consumption, (iii) Brake thermal efficiency, (iv) Volumetric efficiency, and (v) Percentage of excess air supplied.

Solution. $n = 6$, $D = 0.125$ m, $L = 0.125$ m, $N = 2400$ r.p.m.

$W = 490\ N$, C_D = dynamometer constant = 16100

d_0 = orifice diameter = 0.055 m, $C_d = 0.66$, $h_w = 310$ mm

$$\dot{m}_f = \frac{22.1}{3600} = 0.00614 \text{ kg/s}, \; C = 45100 \text{ kJ/kg},$$

$$k = \frac{1}{2} \text{ ..... for 4-stroke cycle engine.}$$

(*i*) **Brake mean effective pressure, p_{mb} :**

Brake power, $$\text{B.P.} = \frac{W \times N}{C_D} = \frac{490 \times 2400}{16100} = 73 \text{ kW}$$

Also $$\text{B.P.} = \frac{n p_{mb} LANk \times 10}{6}$$

$$73 = \frac{6 \times p_{mb} \times 0.125 \times \pi/4 \times 0.125^2 \times 2400 \times \frac{1}{2} \times 10}{6}$$

$\therefore$ $$p_{mb} = \frac{73 \times 6 \times 4 \times 2}{6 \times 0.125 \times \pi \times 0.125^2 \times 2400 \times 10} = 3.96 \text{ bar.} \quad \textbf{(Ans.)}$$

(*ii*) **Specific fuel consumption, b.s.f.c. :**

$$\text{b.s.f.c.} = \frac{22.1}{73} = 0.3027 \text{ kg/kWh.} \quad \textbf{(Ans.)}$$

(*iii*) **Brake thermal efficiency, $\eta_{th.(B)}$:**

$$\eta_{th.(B)} = \frac{\text{B.P.}}{m_f \times C} = \frac{73}{0.00614 \times 45100} = 0.2636 \text{ or } 26.36\%. \quad \textbf{(Ans.)}$$

(*iv*) **Volumetric efficiency, $\eta_{vol.}$:**

Stroke volume of cylinder = $\pi/4\ D^2 \times L$

$= \pi/4 \times 0.125^2 \times 0.125 = 0.00153$ m^3

The volume of air passing through the orifice of the air box per minute is given by,

$$V_a = 840\, A_0\, C_d \sqrt{\frac{h_w}{\rho_a}}$$

where, C_d = Discharge coefficient of orifice = 0.66

A_0 = Area of cross-section of orifice

$= \pi/4\ d_0^2 = \pi/4 \times (0.055)^2 = 0.00237$ m^2

h_w = Head causing flow through orifice in cm of water = $\frac{310}{10} = 31$ cm

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(*ii*) **Brake power, B.P. :**

$$\text{B.P.} = \frac{(W - S) \times D_b N}{60 \times 1000} = \frac{590 \times \pi \times 1 \times 350}{60 \times 1000} = \mathbf{10.81\ kW.} \quad \textbf{(Ans.)}$$

Heat supplied per minute $= \frac{4.3}{60} \times 43900 = 3146$ kJ/min.

(*i*) Heat equivalent of I.P. = 17.38 × 60 = 1042.8 kJ/min.

(*ii*) Heat lost to cooling water

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

$$= \frac{500}{60} \times 4.18 \times (50 - 25) = 870.8 \text{ kJ/min}$$

Now,

$2H_2$	+ O_2	= $2H_2O$
4	32	36
1	8	9

i.e., 1 kg of H_2 produces 9 kg of H_2O

∴ Mass of H_2O produced per kg of fuel burnt

$$= 9 \times H_2 \times \text{mass of fuel used/min.}$$

$$= 9 \times 0.15 \times \frac{4.3}{60} = 0.0967 \text{ kg/min.}$$

Total mass of exhaust gases (wet)/min.

$$= \text{Mass of air/min.} + \text{mass of fuel/min.}$$

$$= \frac{(33 + 1) \times 4.3}{60} = 2.436 \text{ kg/min.}$$

Mass of dry exhaust gases/min.

$$= \text{Mass of wet exhaust gases/min} - \text{mass of } H_2O \text{ produced/min.}$$

$$= 2.436 - 0.0967 = 2.339 \text{ kg/min.}$$

(*iii*) Heat lost to dry exhaust gases

$$= m_g \times c_{pg} \times (t_g - t_r)$$

$$= 2.339 \times 1.0 \times (400 - 25) = 887 \text{ kJ/min.}$$

(*iv*) Assuming that steam in exhaust gases exists as superheated steam at atmospheric pressure and exhaust gas temperature, the enthalpy of 1 kg of steam at atmospheric pressure $1.013 \simeq 1$ bar and 400°C

$$= h_{sup} - h \quad \text{(where } h \text{ is the sensible heat of water at room temperature)}$$

$$= [h_f + h_{fg} + c_{ps}(t_{sup} - t_s)] - 1 \times 4.18 \times (25 - 0)$$

$$= [417.5 + 2257.9 + 2.09\,(400 - 99.6)] - 104.5$$

$$= 3355 \text{ kJ/min.}$$

∴ Heat carried away by steam = 0.0967 × 3355 = 320.6 kJ/min.

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- Major portion of the lead that enters the engine is emitted from the exhaust which forms very small particles of oxides and oxyhalides in the atmosphere. A portion of the lead particles falls to the ground very quickly, others are small enough to remain suspended in the atmosphere sometime, before they fall out, usually after coagulation with other dusty material in air.
- It may not be possible to eliminate lead completely from all petrols immediately because a large number of existing engines rely upon the lubrication provided by a lead film to prevent rapid wear of exhaust valve seats. However, a very small lead content would be adequate for the purpose.

Following points are worth noting :

- Both the *flow rate* and *pollutant concentration*, for exhaust emissions, can change with the mode operation. Both must be considered in determining emissions.
  — *Under constant high speed conditions, exhaust* HC *concentrations are low while the flow rates are high. During accelerations the flow rate is low but* HC *concentration is high.*
  — The *concentration* of HC in the crankcase and evaporative losses is virtually *independent* of operating conditions, but the *flow rates* from each of these sources *change* during various operations.
  Thus, on km basis CO and HC emissions decrease with increasing driving speed while NO_x emissions are relatively not affected.
- In *a poorly maintained engine the exhaust pollution is more.*
  — An automatic choke sticking in the closed position or a very dirty air cleaner element can reduce air-fuel ratio, generally increasing HC and CO emissions.
  — A misfire allows an entire air-fuel charge to be exhausted without combustion.

18.4. S.I. ENGINE EMISSION CONTROL

The main methods, among various methods, for S.I. engine emission control are :

1. Modification in the engine design and operating parameters.
2. Treatment of exhaust products of combustion.
3. Modification of the fuels.

18.4.1. Modification in the Engine Design and Operating Parameters

Engine design modification improves upon the emission quality. A few parameters which improve an emission are discussed below :

1. **Combustion chamber configuration :**

Modification of combustion chamber involves avoiding flame quenching zones where combustion might otherwise be incomplete and resulting in high HC emission. This includes :

— Reduced surface to volume (S/V) ratio ;
— Reduced squish area ;
— Reduced space around piston ring ;
— Reduced distance of the top piston ring from the top of the piston.

2. **Lower compression ratio :**

- Lower compression ratio reduces the quenching effect by reducing the quenching area, thus *reducing HC.*
- Lower compression ratio also *reduces NO_x* emissions due to lower maximum temperature.
- Lower compression, however, *reduces thermal efficiency and increases fuel consumption.*

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- The engine having three lobed rotor is driven eccentrically in a casing in such a way that there are *three separate volumes trapped between the rotor and the casing. These three volumes perform "induction", "compression", "combustion", "expansion" and "exhaust" processes in sequences.* There are three power impulses for each revolution of the rotor, and since the eccentric or output shaft rotates at three times the speed of the rotor, *there is only one power impulse for each revolution of the output shaft of a single bank rotary engine.*

One complete *thermodynamic cycle* is completed over 360° rotation of the rotor ; the suction phase takes 90° of rotor movement and so the also other three phases. One *thermodynamic phase* is completed every 270° rotation of the output shaft, since the output shaft makes three revolution for every single rotation of the rotor.

19.4.3. Features

1. *Simple construction, less mechanical loss, smooth motion and does not require a cranking mechanism.*

2. *Good power volume ratio.*

3. *No reciprocating parts* and hence no balancing problem and complicated engine vibrations eliminated.

4. Due to the absence of intake-exhaust valve mechanism, the *correct timings* for opening and closing (the ports) *can be maintained even at high speeds.*

5. *Low torque fluctuation.*

There are problems in the design, notabty of sealing and of heat transfer but these have been overcome sufficiently well for spark ignition engine to be marketed.

19.4.4. Constructional and Other Details of Wankel Engine

1. Rotor housing and housing materials :

- **Rotor** are generally made from *high-grade malleable spheroidal graphite iron.*
- The **rotor housing** is an *aluminium silicon alloy,* bonded to the cylinder-bore walls in thin sheet metal, the outer surfaces of which have a saw-tooth finish to improve adhesion and thermal conductivity. This lining is then given a hard chromium vanadium plating, which in turn is plated with more chrome but in a thin, porous and oil retaining layer. End and intermediate rotor housings are made from implated high silicon aluminium alloy.
- **Apex and side seal blades** can be made from *cast-iron* but the more popular types are made from *hard carbon material.*
- Both the **leaf and washer springs** can be made from *beryllium* copper, which has the ability to retain its elasticity when operating under working temperatures.

2. Rotor seals :

- The *planetary motion of the rotor within the epitrochoid bore of the rotor housing is designed to maintain a contact between the triangular corner of the rotor and the cylinder walls.* Peripheral radial corner blades, known as the *apex seals,* are necessary to prevent gas leakage between the three cylinder spaces created by the three-sided rotor. Similarly, *side seals* between the flat rotor sides and the end and intermediate housing side walls are essential to stop engine oil reaching the cylinders, and gas from combustion escaping into the eccentric output shaft region.
- The gas-tight sealing between the rotor and housing may be considered in terms of primary and secondary sealing areas.
  - The *primary sealing areas* are those between the sealing elements (apex and side blades) and the cylinder housing bore and side walls.

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Applications :

The following are the *applications of free-piston engines* :

1. Widely used as a submarine air compressor units.
2. Suitable for power generation in medium power range. Below 300 kW diesel engines are indispensable as free piston engine of comparable sizes are not being built on commercial scale.
3. Free piston engines are specially suitable for pumping oil ; also the same oil can be used as fuel.

[illegible]

1. *Duel-fuel operation,* combines in a simple manner the possibility of operating a diesel engine on liquid fuels such as diesel oil or gas oil and on gaseous fuels such as natural gas, sewage gas and cook oven gas etc.
2. At full-load, duel-fuel engine is superior to diesel engine.
3. A *multi-fuel engine* is one which can operate satisfactorily on a wide variety of fuels ranging from diesel oil, crude oil, JP-4 to lighter fuels like gasoline, and even normal lubricating oil.
4. The *stratified charge engine* is usually defined as a S.I. engine (stratified diesel engine has also been developed) in which the mixture in the zone of spark plug is very much richer than that in the rest of the combustion chamber *i.e.* one which burns leaner overall fuel-air mixtures. *Charge stratification* means providing different fuel-air mixture strengths at various places in the combustion chamber.
5. The basic principle of working of *stirling engine* is the same as that of conventional engine. *The alternate compression at low temperature and expansion at high temperature of a working fluid in the basis for the stirling engine.*
6. In the *VCR-engine* a high compression is used for good stability and low load operation and a low compression is used at full-load to allow the turbocharger to boost the intake pressures without increasing the peak cycle pressure.
7. *Free-piston engine plants* are the conventional gas turbine plants with the *difference that the air compressor and combustion chamber are replaced by free-piston engine.*

OBJECTIVE TYPE QUESTIONS

Fill in the Blanks or Say "Yes" or "No" :

1. A ...... engine is capable of running on either gas or diesel oil or a combination of these two over a wide range of temperature ratio.
2. The use of low octane number fuels in duel-fuel engine results in poor performance of the engine and greatly affects the combustion.
3. In a duel-fuel engine the temperature of inlet charge has no effect on the knocking limits of a particular fuel-air mixture.
4. At full-load, duel-fuel engine is superior to diesel engine.
5. A duel-fuel engine is preferred when cheap gas is easily available.
6. A ...... engine is one which can operate satisfactorily on a wide variety of fuels.
7. ...... means providing different fuel-air mixture strengths at various places in the combustion chamber
8. The stratified charge engine combines the advantages of both petrol and diesel engines.
9. A stratified charge engine exhibits multi-fuel capability.
10. The stirling engine is an external combustion engine.
11. The part-load efficiency of a stirling engine is very low.
12. In case of stirling engine no lubricating oil is required.

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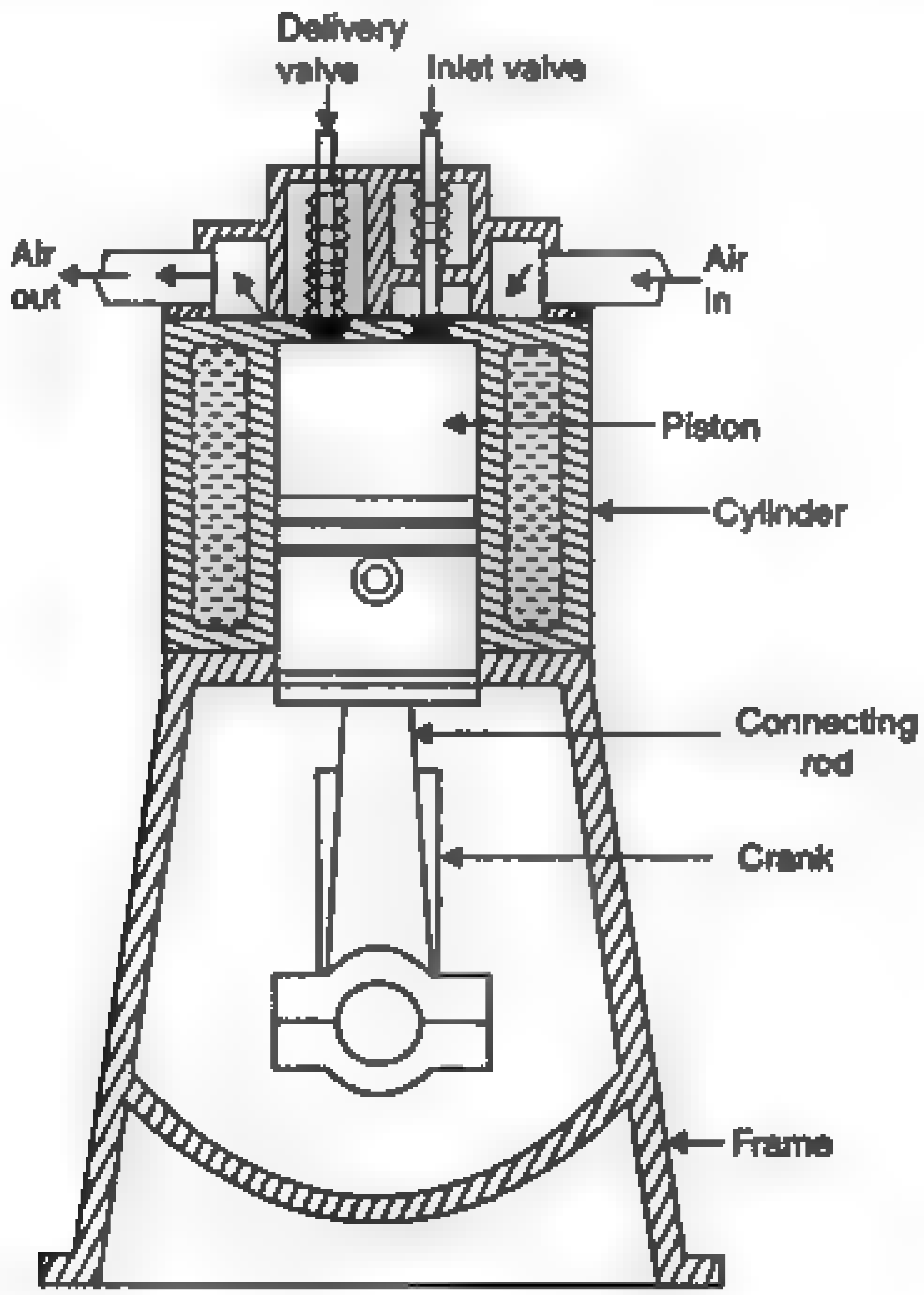
— *Blowers* and *superchargers* are essentially air compressors, but the *increase in pressure* which they produce is only small, and upto, say *0.7 to 1 05 bar.*

— *A booster is an air or gas compressor which is employed to raise the pressure of air / gas which has already been compressed.* It is where a slightly higher pressure is required, or where a loss of pressure has occurred in a long delivery line.

20.3. RECIPROCATING COMPRESSORS

20.3.1. Construction and Working of a Reciprocating Compressor (Single-stage)

Fig. 20.2 (*a*) shows a sectional view of a single-stage reciprocating compressor. It consists of a piston which reciprocates in a cylinder, driven through a connecting rod and crank mounted in a crankcase. There are inlet and delivery valves mounted in the head of the cylinder. These valves are usually of the pressure differential type, meaning that they will operate as the result of the difference of pressures across the valve. The working of this type of compressor is as follows :



(*a*)

Fig. 20.2 (*a*) Sectional view of a single-stage reciprocating compressor.

As shown in Fig. 20.2 (*b*), the piston is moving down the cylinder and any residual compressed air left in the cylinder after the previous compression will expand and will eventually

$$p_4 = Zp_3 = Z^3 p_1$$

$$\vdots$$

$$p_{x+1} = Zp_x = Z^x p_1$$

$\therefore$
$$Z^x = \frac{p_{x+1}}{p_1}$$

or
$$Z = \sqrt[x]{\frac{p_{(x+1)}}{p_1}} = x\ \sqrt{\text{(Pressure ratio through compressor)}} \quad ...(20.34)$$

Inserting the value of Z in eqn. (20.33) will determine the intermediate pressures.

In the event of *intercooling being imperfect we must treat each stage as a separate compressor*, in which case 'x' in eqn. (20.31) will be unity. With this special value of 'x' the power per stage can be calculated, and finally the total power is the sum of the powers per stage :

$$W = \frac{n_1}{n_1 - 1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_1 - 1}{n_1}} - 1 \right] + \frac{n_2}{n_2 - 1} p_2 V_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n_2 - 1}{n_2}} - 1 \right] + ...$$

Heat rejection per stage per kg of air :

If the air is cooled to its initial temperature the whole of the work done in compression must be rejected to the cooling medium.

Hence for a *single-stage* the heat rejected is given by,

Heat rejected
$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad ...(20.35)$$

and since for 1 kg of air, $p_1V_1 = RT_1$ and $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}}$, then eqn. (20.35) may be written as

$$W = \frac{n}{n-1} RT_1 \left[\frac{T_2}{T_1} - 1 \right] \text{ per kg of air}$$

$$= \frac{n}{n-1} \frac{R}{J} (T_2 - T_1) \text{ heat units} \qquad \begin{bmatrix} J = 1 \text{......S.I. units} \\ J = 427 \text{...... M.K.S. units} \end{bmatrix}$$

But
$$\frac{R}{J} = c_p - c_v$$

$$W = \frac{n}{n-1} (c_p - c_v)(T_2 - T_1) \quad ...(20.36)$$

$\therefore$ ***Heat rejected with perfect intercooling***

$$= \left[c_p + c_v \left(\frac{\gamma - n}{n - 1} \right) \right] (T_2 - T_1) \text{ per kg of air} \quad ...(20.37)$$

$$\left[\textbf{Note. } \frac{n}{n-1} (c_p - c_v) = c_p + \frac{c_v (\gamma - n)}{n - 1} \right]$$

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The sliding blade eccentric drum type requires internal lubrication, and even so the slots, in which the blades move, wear rapidly.

The toothed wheel type has a smaller friction and can expand damp air without internal deterioration. In the "herringbone" type expansion is possible together with a high starting torque and extreme mechanical simplicity. This commends the turbine for colliery work inspite of its extravagance on air.

Example 20.1. *A single-stage reciprocating compressor takes 1 m^3 of air per minute at 1.013 bar and 15°C and delivers it at 7 bar. Assuming that the law of compression is $pV^{1.35}$ = constant, and the clearance is negligible, calculate the indicated power.*

Solution. Volume of air taken in, $V_1 = 1$ m^3/min

Intake pressure, $p_1 = 1.013$ bar

Initial temperature, $T_1 = 15 + 273 = 288$ K

Delivery pressure, $p_2 = 7$ bar

Law of compression : $pV^{1.35}$ = constant

Indicated power I.P. :

Mass of air delivered per min.,

$$m = \frac{p_1V_1}{RT_1} = \frac{1.013 \times 10^5 \times 1}{287 \times 288} = 1.226 \text{ kg/min}$$

Delivery temperature, $$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{(n-1)/n}$$

$$= 288 \left(\frac{7}{1.013}\right)^{(1.35-1)/1.35} = 475.2 \text{ K}$$

Indicated work $$= \frac{n}{n-1} \, mR\,(T_2 - T_1) \text{ kJ/min}$$

$$= \frac{1.35}{1.35-1} \times 1.226 \times 0.287\,(475.2 - 288) = 254 \text{ kJ/min}$$

i.e., *Indicated power I.P* $= \frac{254}{60} =$ **4.23 kW. (Ans.)**

Example 20.2. *If the compressor of example 21.1 is driven at 300 r.p.m. and is a single-acting, single-cylinder machine, calculate the cylinder bore required, assuming a stroke to bore ratio of 1.5 : 1. Calculate the power of the motor required to drive the compressor if the mechanical efficiency of the compressor is 85% and that of the motor transmission is 90%.*

Solution. Volume dealt with per minute at inlet = 1 m^3/min.

$\therefore$ Volume drawn in per cycle $= \frac{1}{300} = 0.00333$ m^3/cycle

i.e., Cylinder volume $= 0.00333$ m^3

$$\therefore \quad \frac{\pi}{4} D^2L = 0.00333$$

(where D = bore, L = stroke)

i.e., $$\frac{\pi}{4} D^2\,(1.5 \times D) = 0.00333 \quad \text{or} \quad D^3 = \frac{0.00333 \times 4}{\pi \times 1.5}$$

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Example 20.5. *(a) Show that the compressor work obtained from the analysis of a conventional card with clearance and polytropic processes for the reciprocating compressor is identical to that obtained from the analysis of a reversible steady flow rotary compressor where in certain mass of a gas is compressed from the initial condition of pressure p_1 and t_1 respectively to the final pressure p_2 in accordance to $pv^n = C$.*

(b) A low pressure, water jacketed steady flow rotary compressor compresses polytropically 6.75 kg/min of air from 1 atm. and 21°C to 0.35 bar gauge and 43°C. Neglecting the change in kinetic energy find the work and mass of water circulated if the temperature rise of the cooling water is 3.3°C. Take c_p (for air) = 1.003 kJ/kg K. **(P.U.)**

Solution. (*a*) In reciprocating compressors the work required is

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{n}{n-1} mR\,(T_2 - T_1) \qquad ...(1)$$

When compression is adiabatic, $n = \gamma$

Work transfer in rotary compressor is determined by *applying steady flow energy equation.*

(*i*) **With isentropic flow.** Applying steady flow energy equation, we get

$$h_1 + \frac{C_1^2}{2g} + W = h_2' + \frac{C_2^2}{2g}$$

For $C_1 = C_2$, $W = h_2' - h_1 = c_p(T_2' - T_1)$

where T_2' is the temperature after isentropic compression

Since $\qquad c_p = \dfrac{\gamma R}{\gamma - 1}$

$$W = \frac{\gamma}{\gamma - 1} R(T_2' - T_1) \qquad ...(2)$$

Eqn. (1) is similar to eqn. (2) for unit mass.

(*ii*) **With compression polytropic.** In actual practice due to internal heating there is increase of work done above isentropic work, and work done is

$$W = c_p(T_2 - T_1) = \frac{\gamma}{\gamma - 1} R(T_2 - T_1)$$

where T_2 is the actual temperature, *i.e.*, obtained by using the relationship $pv^n = C$.

(*iii*) **With cooled compression.** Some heat is being taken away by cooling of compressor and so

$$W = c_p(T_2 - T_1) + Q$$

(*b*) **Work,** $\qquad W = m\,c_p\,(T_2 - T_1)$

$= 6.75 \times 1.003\,(43 - 21) =$ **148.94 min. (Ans.)**

If the compression would have been isentropic

$$T_2' = T_1\,(r_p)^{\frac{\gamma-1}{\gamma}} = (21 + 273)\left(\frac{1.35}{1}\right)^{\frac{1.4-1}{1.4}} = 320.3\text{°K} \quad \text{or} \quad 47.3\text{°C}$$

Heat rejected to cooling water

$= m\,c_p(T_2' - T_2)$

$= 6.75 \times 1.003\,(47.3 - 43) = 29.11$ kJ

Mass of cooling water, $m_w = \dfrac{29.11}{c_{pw} \times (t_{w2} - t_{w1})} = \dfrac{29.11}{4.18 \times 3.3} =$ **2.11 kg/min. (Ans.)**

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Example 20.11. *A 4-cylinder double-acting compressor is required to compress 30 m^3/min. of air at 1 bar and 27°C to a pressure of 16 bar. Determine the size of motor required and cylinder dimensions if the following data is given :*

Speed of the compressor,	$N = 320$ r.p.m.
Clearance volume,	$V_c = 4\%$
Stroke to bore ratio,	$L/D = 1.2$
Mechanical efficiency,	$\eta_{mech.} = 82\%$
Value of index,	$n = 1.32.$

Assume no pressure change in suction valves and the air gets heated by 12°C during suction stroke.

Solution. Refer Fig. 20.11.

$$\text{Net work done} = \frac{n}{n-1} \times p_1 \times (V_1 - V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

where $V_1 - V_4$ = suction volume = 30 m³/min (given) = $\frac{30}{60}$ = 0.5 m³/s

$$\therefore \quad \text{Work done} = \frac{1.32}{1.32-1} \times 1 \times 10^5 \times 0.5\left[\left(\frac{16}{1}\right)^{\frac{1.32-1}{1.32}} - 1\right] = 197648.9 \text{ Nm/s}$$

$$\text{Theoretical power} = \frac{197648.9}{1000} = 197.64 \text{ kW}$$

$$\therefore \quad \textbf{Motor power} = \frac{197.64}{\eta_{mech.}} = \frac{197.64}{0.82} = \mathbf{241\ kW} \quad \textbf{(Ans.)}$$

$$\text{Volumetric efficiency, } \eta_{vol.} = \left[1 + k - k\left(\frac{p_2}{p_1}\right)^{1/n}\right] \times \frac{p_i T_a}{p_a T_i}$$

(Suffix 'i' and 'a' stand for inside and atmospheric conditions)

i.e.,
$$\eta_{vol.} = \left[1 + 0.04 - 0.04\left(\frac{16}{1}\right)^{1/1.32}\right] \times \frac{1 \times (273+27)}{1 \times (273+39)} = 0.686 \text{ or } 68.6\%$$

Now swept volume of one cylinder

$$= \frac{30}{4} \times \frac{1}{2 \times 320} \times \frac{1}{0.686} = 0.01708 \text{ m}^3$$

$$\therefore \quad \frac{\pi}{4} D^2 L = 0.01708 \quad \text{or} \quad \frac{\pi}{4} D^2 \times 1.2\, D = 0.01708$$

$$\therefore \quad D^3 = \frac{0.01708 \times 4}{\pi \times 1.2}$$

$$\therefore \quad \mathbf{D = 0.263\ m \quad or \quad 263\ mm.} \quad \textbf{(Ans.)}$$

and
$$\mathbf{L = 1.2 \times 263 = 315.6\ mm.} \quad \textbf{(Ans.)}$$

Example 20.12. *A two-cylinder single-acting air compressor is to deliver 16 kg of air per minute at 7 bar from suction conditions 1 bar and 15°C. Clearance may be taken as 4% of stroke volume and the index for both compression and re-expansion as 1.3. Compressor is directly*

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Compression, expansion index, $n = 1.3$

Clearance volume in each stage $= 5\%$ of swept volume

Speed of the compressor, $N = 300$ r.p.m.

(*i*) **Indicated power :**

$$\therefore \quad \frac{p_i}{p_s} = \frac{p_d}{p_i}$$

$$\therefore \quad p_i^2 = p_s \times p_d = p_s \times 9p_s = 9p_s^2$$

$$\therefore \quad p_i = 3\,p_s \quad i.e., \quad \frac{p_i}{p_s} = 3$$

Now using the equation, $\dfrac{T_i}{T_s} = \left(\dfrac{p_i}{p_s}\right)^{\frac{n-1}{n}} = (3)^{\frac{1.3-1}{1.3}}$

$$\therefore \quad T_i = T_s \times (3)^{0.3/1.3} = 288 \times (3)^{0.3/1.3} = 371 \text{ K}$$

Now as *n*, *m* and temperature difference are the same for both stages, then the *work done in each stage is the same.*

Total work required per min. $= 2 \times \dfrac{n}{n-1}\, mR\,(T_i - T_s)$

$$= 2 \times \frac{1.3}{1.3-1} \times 4.5 \times 0.287\,(371 - 288) = 929 \text{ kJ/min.}$$

$\therefore$ **Indicated power** $= \dfrac{[redacted]}{60} = $ **15.48 kW. (Ans.)**

(*ii*) **The cylinder swept volumes required :**

The mass induced per cycle, $m = \dfrac{4.5}{300} = 0.015$ kg/cycle.

This mass is passed through each stage in turn

For the L.P. pressure cylinder (Fig. 20.30)

$$V_1 - V_4 = \frac{mRT_s}{p_s} = \frac{0.015 \times 287 \times 288}{1.013 \times 10^5} = 0.0122 \text{ m}^3\text{/cycle}$$

$$\eta_{vol.} = \frac{V_1 - V_4}{V_s} = 1 + k - k\left(\frac{p_i}{p_s}\right)^{1/n} = 1 + 0.05 - 0.05\,(3)^{1/1.3}$$

$$= 0.934 \qquad \left(k = \frac{V_c}{V_s} = 0.05\right)$$

i.e., $\eta_{vol.} = 0.934$

$$\therefore \quad V_s = \frac{V_1 - V_4}{\eta_{vol.}} = \frac{V_1 - V_4}{0.934} = \frac{0.0122}{0.934} = 0.0131 \text{ m}^3\text{/cycle}$$

i.e., **Swept volume of L.P. cylinder $V_{s(L.P.)} = 0.0131$ m³. (Ans.)**

For the high pressure stage, a mass of 0.015 kg/cycle is drawn in at 15°C and a pressure of $p_i = 3 \times 1.013 = 3.039$ bar

i.e., Volume drawn in $= \dfrac{0.015 \times 287 \times 288}{3.039 \times 10^5} = 0.00408$ m³/cycle

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UNSOLVED EXAMPLES

1. Air is to be compressed in a single-stage reciprocating compressor from 1.013 bar and 15°C to 7 bar. Calculate the indicated power required for a free air delivery of 0.3 m^3/min, when the compression process is
   (i) Isentropic (ii) Reversible isothermal
   (iii) Polytropic, with n = 1.25. [**Ans.** 1.51 kW ; 0.98 kW ; 1.19 kW]
2. The compressor of the above example is to run at 1000 r.p.m. If the compressor is single-acting and has a stroke/bore ratio of 1.2/1, calculate the bore size required. [**Ans.** 68.3 mm]
3. A single-stage, single-acting air compressor running at 1000 r.p.m. delivers air at 25 bar. For this purpose the induction and free air conditions can be taken as 1.013 bar and 15°C, and the free air delivery as 0.25 m^3/min. The clearance volume is 3% of the swept volume and the stroke/bore ratio is 1.2 : 1. Calculate the bore and stroke and the volumetric efficiency of this machine. Take the index of compression and expansion as 1.3. Calculate also the indicated power and the isothermal efficiency.
   [**Ans.** 73.2 mm ; 87.84 mm ; 67.6% ; 2 kW ; 67.5%]
4. A single acting compressor is required to deliver air at 70 bar from an induction pressure of 1 bar, at the rate of 2.4 m^3/min measured at free-air conditions of 1.013 bar and 15°C. The temperature at the end of the induction stroke is 32°C. Calculate the indicated power required if the compression is carried out in two stages with an ideal intermediate pressure and complete intercooling. The index of compression and expansion for both stages is 1.25. What is the saving in power over single-stage compression ?
   If the clearance volume is 3% of the swept volume in each cylinder, calculate the swept volumes of the cylinders. The speed of the compressor is 750 r.p.m.
   If the mechanical efficiency of the compressor is 85%, calculate the power output in kilowatts of the motor required. [**Ans.** 22.7 kW ; 6 kW ; 0.00396 m^3, 0.000474 m^3, 26.75 kW]
5. A single-cylinder, single-acting air compressor running at 300 r.p.m. is driven by a 23 kW electric motor. The mechanical efficiency of the drive between motor and compressor is 87%. The air inlet conditions are 1.013 bar and 15°C and the delivery pressure is 8 bar. Calculate the free-air delivery in m^3/min, the volumetric efficiency, and the bore and stroke of the compressor. Assume that the index of compression and expansion is n = 1.3, that the clearance volume is 7% of the swept volume and that the bore is equal to the stroke. [**Ans.** 4.47 m^3/min ; 73% ; 296 mm]
6. A two-stage air compressor consists of three cylinders having the same bore and stroke. The delivery pressure is 7 bar and the free air delivery is 4.2 m^3/min. Air is drawn in at 1.013 bar, 15°C and an intercooler cools the air to 38°C. The index of compression is 1.3 for all the three cylinders. Neglecting clearance calculate
   (i) The intermediate pressure (ii) The power required to drive the compressor
   (iii) The isothermal efficiency [**Ans.** (i) 2.19 bar (ii) 16.3 kW (iii) 84.5%]
7. A two-stage double-acting air compressor, operating at 200 r.p.m., takes in air at 1.013 bar and 27°C. The size of the L.P. cylinder is 350 × 380 mm ; the stroke of H.P. cylinder is the same as that of the L.P. cylinder and the clearance of both the cylinders is 4%. The L.P. cylinder discharges the air at a pressure of 4.052 bar. The air passes through the intercooler so that it enters the H.P. cylinder at 27°C and 3.85 bar, finally it is discharged from the compressor at 15.4 bar. The value of 'n' in both cylinders is 1.3. c_p = 1.0035 kJ/kg K and R = 0.287 kJ/kg K.
   Calculate : (i) The heat rejected in the intercooler
   (ii) The diameter of H.P. cylinder (iii) The power required to drive the H.P. cylinder
   [**Ans.** (i) 1805.68 kg/min. (ii) 179.5 mm (iii) 37.3 kW]
8. A single-acting two-stage compressor with complete intercooling delivers 10 kg/min of air at 16 bar. The suction occurs at 1 bar and 15°C. The expansion and compression processes are reversible polytropic with polytropic index n = 1.25. Calculate :
   (i) The power required. (ii) The thermal efficiency
   (iii) The free air delivery (iv) Heat transferred in intercooler

(ii) **Merits over steam turbines :**

The gas turbine entails the following *advantages over steam turbines* :

1. Capital and running cost less.
2. For the same output the space required is far less.
3. Starting is more easy and quick.
4. Weight per H.P. is far less.
5. Can be installed anywhere.
6. Control of gas turbine is much easier.
7. Boiler along with accessories not required.

21.4. CONSTANT PRESSURE COMBUSTION GAS TURBINES

21.4.1. Open Cycle Gas Turbines

Refer Fig. 21.1. The fundamental gas turbine unit is one operating on the open cycle in which a rotary compressor and a turbine are mounted on a common shaft. Air is drawn into the compressor and after compression passes to a combustion chamber. Energy is supplied in the combustion chamber by spraying fuel into the air stream, and the resulting hot gases expand through the turbine to the atmosphere. In order to achieve net work output from the unit, the turbine must develop more gross work output than is required to drive the compressor and to overcome mechanical losses in the drive. The products of combustion coming out from the turbine are exhausted to the atmosphere as they cannot be used any more. The working fluids (air and fuel) must be replaced continuously as they are exhausted into the atmosphere.

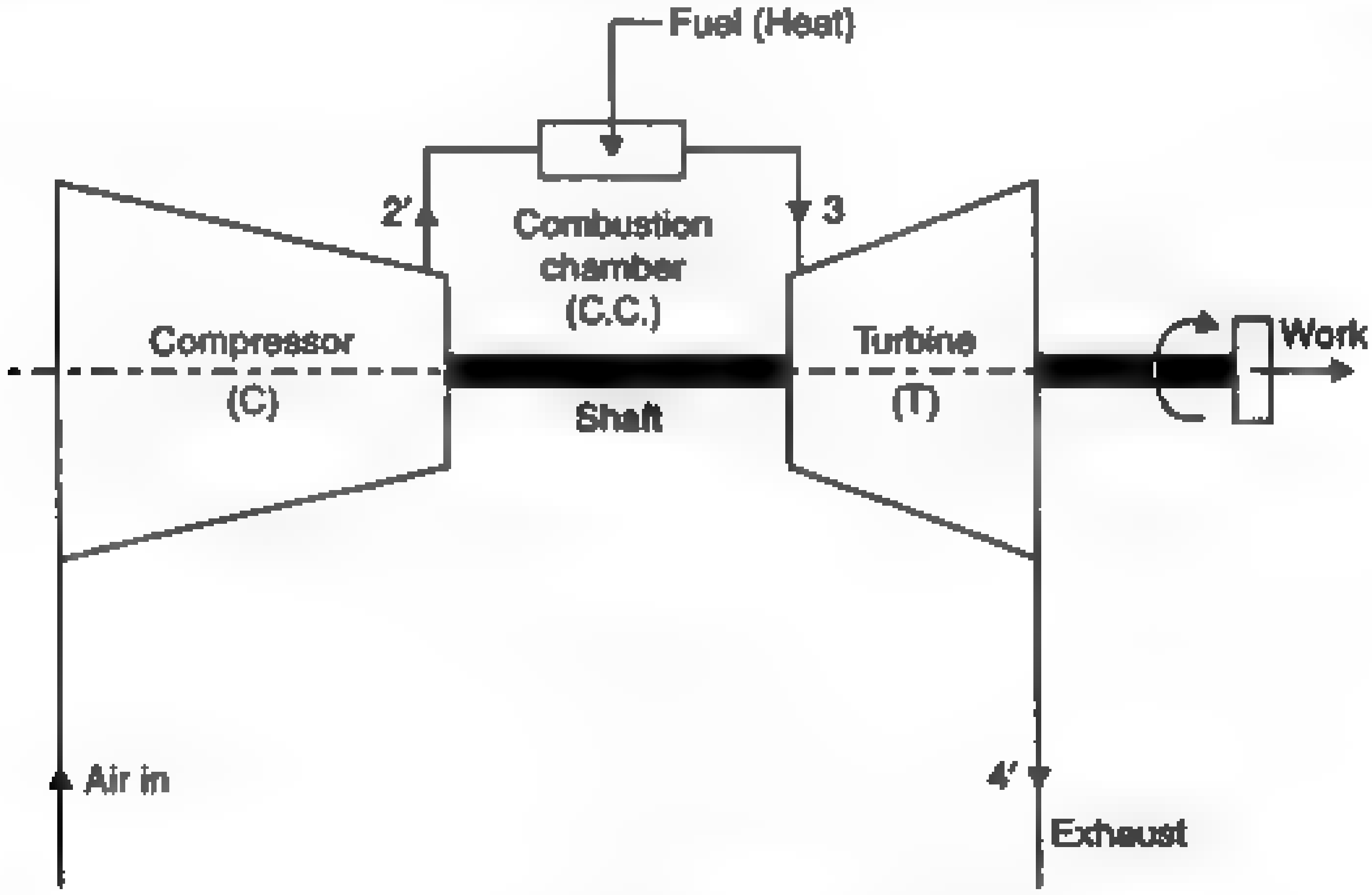

Fig. 21.1. Open cycle gas turbine.

If pressure loss in the combustion chamber is neglected, this cycle may be drawn on a *T-s* diagram as shown in Fig. 21.2.

- 1-2′ represents : *irreversible adiabatic compression.*
- 2′-3 represents : *constant pressure heat supply in the combustion chamber.*
- 3-4′ represents : *irreversible adiabatic expansion.*

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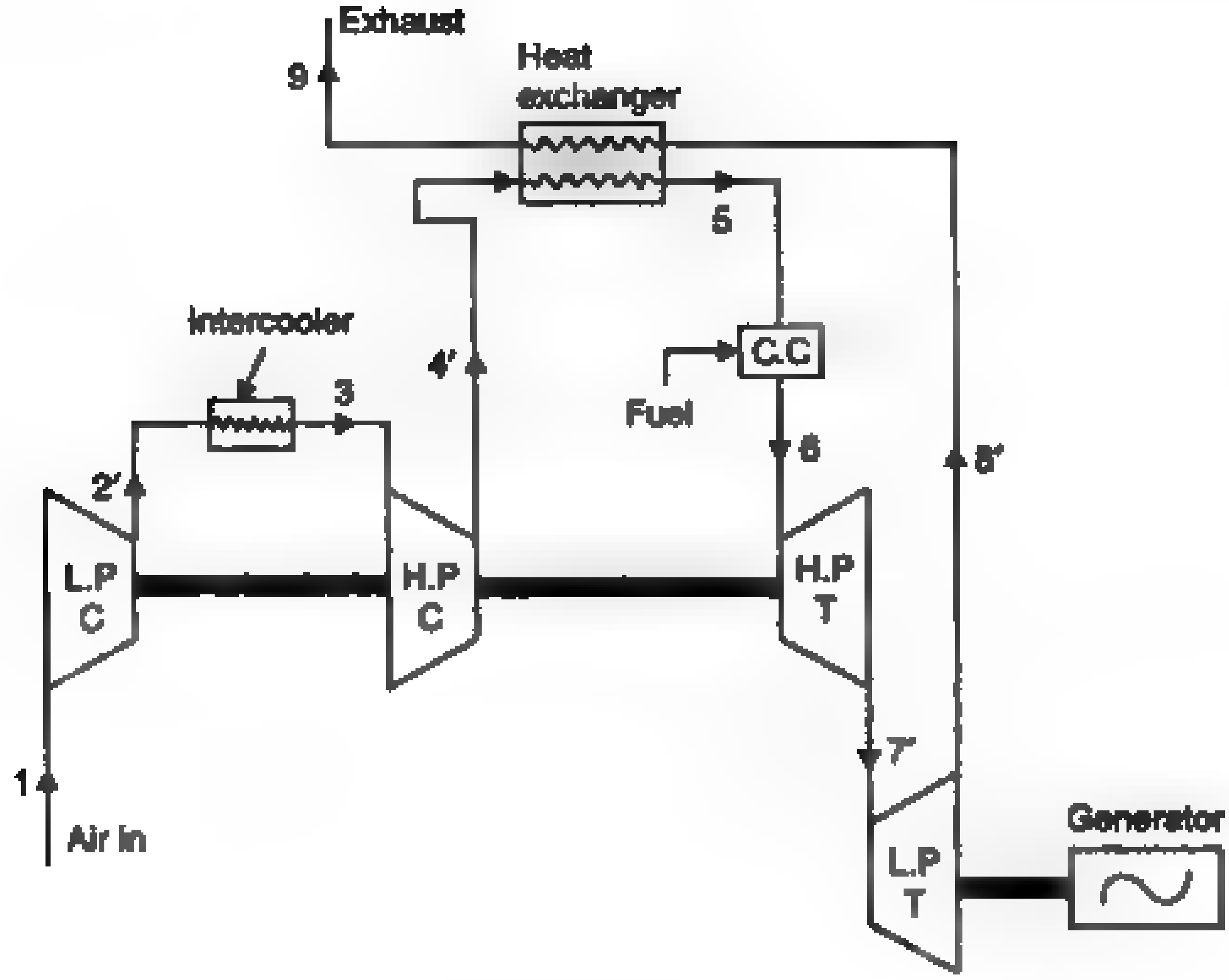
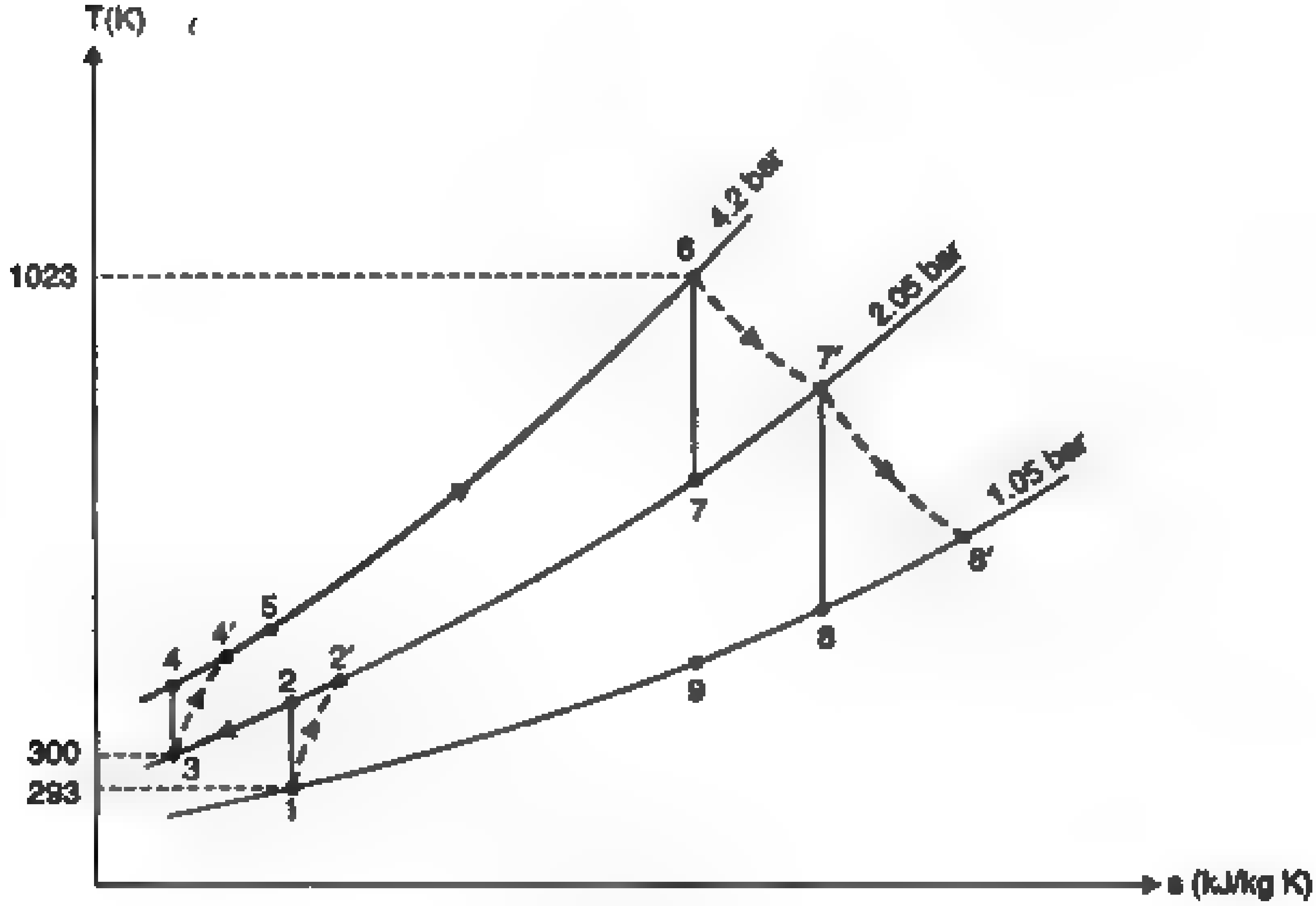


Exhaust
9
Heat
exchanger
5
Intercooler
4′
3
Fuel
C.C
6
8′
2′
L.P
C
H.P
C
H.P
T
7′
1
Air in
Generator
L.P
T

(a)



T(K)
1023
300
293
4.2 bar
2.05 bar
1.05 bar
6
7′
7
8′
8
9
5
4′
4
2
2′
3
1
s (kJ/kg K)

(b)

Fig. 21.35

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6. Rate of climb higher.
7. Requirement of major overhauls less frequent.
8. Radio interference much less.
9. Maximum altitude ceiling as compared to turbo-prop and conventional piston type engines.
10. Frontal area smaller.
11. Fuel can be burnt over a large range of mixture strength.

Disadvantages of turbo-jet engines

1. Less efficient.
2. Life of the unit comparatively shorter.
3. The turbo-jet becomes rapidly inefficient below 550 km/h.
4. More noisy (than a reciprocating engine).
5. Materials required are quite expensive.
6. Require longer strip since length of take-off is too much.
7. At take-off the thrust is low, this effect is overcome by boosting.

21.8.1.2. Basic Cycle for Turbo-jet Engine

The basic cycle for the turbo-jet engine is the *Joule or Brayton cycle* as shown is Fig. 21.39 The various processes are as follows :

Process 1-2 : The air entering from atmosphere is *diffused isentropically* from velocity C_1 down to zero (*i.e.*, $C_2 = 0$). This indicates that the diffuser has an efficiency of 100%, this is termed as *ram compression*.

Process 1-2′ is the *actual process*.

Process 2-3 : *Isentropic* compression of air.

Process 2′-3′ shows the *actual* compression of air.

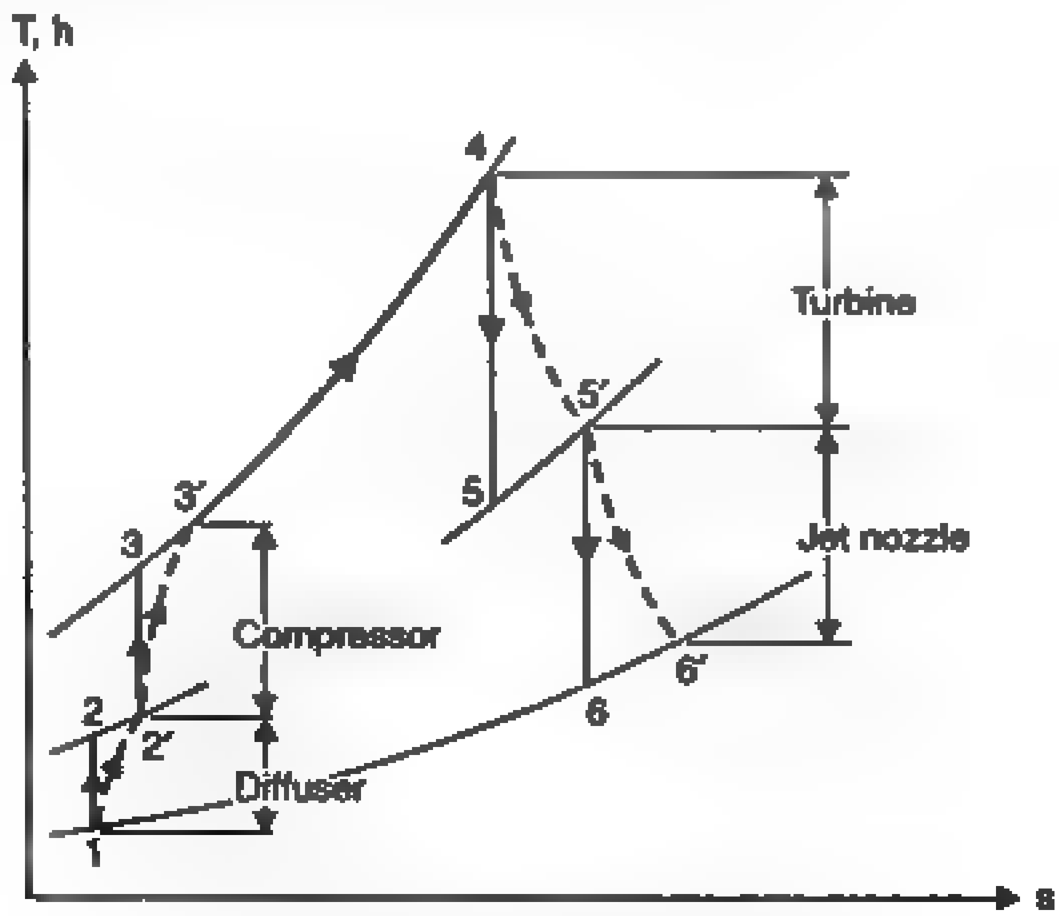

Fig. 21.39. *T-s* diagram of turbo-jet.

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(*vi*) **Propulsive efficiency, $\eta_{prop.}$:**

$$\eta_{prop.} = \frac{\text{Thrust power}}{\text{Propulsive power}} = \frac{2C_a}{C_j + C_a} \quad ...(21.18)$$

$$= \frac{2 \times 277.8}{651 + 277.8} = 0.598 \text{ or } 59.8\%. \quad \textbf{(Ans.)}$$

(*vii*) **Overall efficiency, η_0 :**

$$\eta_0 = \frac{\text{Thrust work}}{\text{Heat supplied by fuel}} = \frac{(C_j - C_a)C_a}{\left(\frac{m_f}{m_a}\right) \times \text{C.V.} \times \eta_{combustion}} \quad ...(21.22)$$

$$= \frac{(651 - 277.8) \times 277.8}{\frac{1}{70} \times 42000 \times 0.92 \times 1000} = 0.1878 \text{ or } 18.78\%. \quad \textbf{(Ans.)}$$

Example 21.20. *The following data pertain to a turbo-jet flying at an altitude of 9500 m :*

Speed of the turbo-jet	*= 800 km/h*
Propulsive efficiency	*= 55%*
Overall efficiency of the turbine plant	*= 17%*
Density of air at 9500 m altitude	*= 0.17 kg/m³*
Drag on the plane	*= 6100 N*

Assuming calorific value of the fuels used as 46000 kJ/kg,

Calculate :

(*i*) *Absolute velocity of the jet.* (*ii*) *Volume of air compressed per min.*
(*iii*) *Diameter of the jet.* (*iv*) *Power output of the unit.*
(*v*) *Air-fuel ratio.*

Solution. *Given :* Altitude = 9500 m, $C_a = \frac{800 \times 1000}{60 \times 60} = 222.2$ m/s,

$\eta_{propulsive} = 55\%$, $\eta_{overall} = 17\%$; density of air at 9500 m altitude = 0.17 kg/m^3 ; drag on the plane = 6100 N.

(*i*) **Absolute velocity of the jet, $(C_j - C_a)$:**

$$\eta_{propulsive} = 0.55 = \frac{2C_a}{C_j + C_a}$$

where, C_j = Velocity of gases at nozzle exit relative to the aircraft, and
C_a = Velocity of the turbo-jet/air-craft.

$$\therefore \quad 0.55 = \frac{2 \times 222.2}{C_j + 222.2}$$

i.e.,

$$C_j = \frac{2 \times 222.2}{0.55} - 222.2 = 585.8 \text{ m/s}$$

$\therefore$ Absolute velocity of jet $= C_j - C_a = 585.8 - 222.2 = 363.6$ m/s.

(*ii*) **Volume of air compressed/min. :**

Propulsive force $= m_a (C_j - C_a)$

$$6100 = m_a (585.8 - 222.2)$$

$$\therefore \quad m_a = 16.77 \text{ kg/s}$$

$\therefore$ Volume of air compressed/min. $= \frac{16.77}{0.17} \times 60 = $ **5918.8 kg/min.** **(Ans.)**

and $$\eta_n = \frac{T_5' - T_6'}{T_5' - T_6} \quad \text{or} \quad T_6' = T_5' - \eta_n (T_5' - T_6)$$
$$= 1171.8 - 0.9(1171.8 - 813.75) = 849.5 \text{ K}$$

Velocity at the exit of the nozzle,

$$C_j = 44.72\sqrt{h_5' - h_6'} = 44.72\sqrt{c_p (T_5' - T_6')}$$
$$= 44.72\sqrt{1.005(1171.8 - 849.5)} = 804.8 \text{ m/s}$$

Specific thrust $$= (1 + m_f) \times C_j = \left(1 + \frac{1}{48.34}\right) \times 804.8$$
$$= \mathbf{821.45 \text{ N/kg of air/s.}} \quad \textbf{(Ans.)}$$

(*iii*) **Total Thrust :**

Volume of flowing air, $V_1 = 0.12 \times 216 = 25.92 \text{ m}^3/\text{s}$

Mass flow, $$m_a = \frac{p_1 V_1}{RT_1} = \frac{0.78 \times 10^5 \times 25.92}{(0.287 \times 1000) \times 265.8} = 26.5 \text{ kg/s}$$

∴ Total thrust $= 26.5 \times 821.45 = \mathbf{21768.4 \text{ N.}}$ **(Ans.)**

Example 21.28. *The following data pertain to a jet engine flying at an altitude of 9000 metres with a speed of 215 m/s.*

Thrust power developed	*750 kW*
Inlet pressure and temperature	*0.32 bar, – 42°C*
Temperature of gases leaving the combustion chamber	*690°C*
Pressure ratio	*5.2*
Calorific value of fuel	*42500 kJ/kg*
Velocity in ducts (constant)	*195 m/s*
Internal efficiency of turbine	*86%*
Efficiency of compressor	*86%*
Efficiency of jet tube	*90%*

For air : $c_p = 1.005$, $\gamma = 1.4$, $R = 0.287$

For combustion gases, $c_p = 1.087$

For gases during expansion, $\gamma = 1.33$.

Calculate the following :

(*i*) *Overall thermal efficiency of the unit ;*

(*ii*) *Rate of air consumption ;*

(*iii*) *Power developed by the turbine ;*

(*iv*) *The outlet area of jet tube ;*

(*v*) *Specific fuel consumption is kg per kg of thrust.*

Solution. Refer Fig. 21.42.

Given : T.P. = 750 kW ; $p_1 = 0.32$ bar, $T_1 = -42 + 273 = 231$ K ; $T_3 = 690 + 273 = 963$ K , $r_{pc} = 5.2$; $C = 42500$ kJ/kg ; $C_a = 215$ m/s, $C_4' = 19.5$ m/s, $\eta_c = 0.86$; $\eta_t = 0.86$; $\eta_{jt} = 0.9$.

Refer Fig. 21.42.

Let m_f = kg of fuel required per kg of air

Then, heat supplied per kg of air

$$= 42500\, m_f = (1 + m_f) \times 1.087(T_3 - T_2') \qquad ...(i)$$

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2. Lethal weapons
3. Signalling and firework display
4. Jet assisted take-off
5. For satellites
6. For space ships
7. Research.

21.8.5.3. Thrust work, propulsive work and propulsive efficiency

In rocket propulsion, since air is self contained, the entry velocity relative to aircraft is zero. Neglecting the friction and other losses, we have the following formulae.

Thrust work $= C_j C_a$

Propulsive work $= C_j C_a + \dfrac{(C_j - C_a)^2}{2} = \dfrac{C_j^2 + C_a^2}{2}$

Rocket propulsive efficiency $$= \frac{C_j C_a}{(C_j^2 + C_a^2)/2} = \frac{2 C_j C_a}{C_j^2 + C_a^2} = \frac{2\left(\dfrac{C_a}{C_j}\right)}{1 + \left(\dfrac{C_a}{C_j}\right)^2} \quad ...(21.23)$$

HIGHLIGHTS

1. The gas turbines are mainly divided into two groups :
   (i) Constant pressure combustion gas turbine
      (a) Open cycle constant pressure gas turbine
      (b) Closed cycle constant pressure gas turbine.
   (ii) Constant volume combustion gas turbine.
2. Methods for improvement of thermal efficiency of open cycle gas turbine plant :
   (i) Intercooling (ii) Reheating (iii) Regeneration.
3. Types of jet propulsion systems :
   (i) Screw propeller (ii) Turbo-jet
   (iii) Turbo-prop (iv) Ram-jet.
4. Difference between jet propulsion and rocket propulsion :
   The main difference is that in case of jet propulsion the oxygen required for combustion is taken from the atmosphere and fuel is stored whereas for rocket engine the fuel and oxidiser both are contained in a propelling body and as such it can function in vacuum.
5. Classification of rockets :
   (i) According to the type of propellents :
      (a) Solid propellent rocket (b) Liquid propellent rocket.
   (ii) According to the number of motors :
      (a) Single-stage rocket (consists of one rocket motor)
      (b) Multi-stage rocket (consists of more than one rocket motor).

OBJECTIVE TYPE QUESTIONS

Choose the Correct Answer

1. Thermal efficiency of a gas turbine plant as compared to Diesel engine plant is
   (a) higher (b) lower
   (c) same (d) may be higher or lower.

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15. The specific gravity of petrol is
(*a*) 1 (*b*) 0.82 (*c*) 0.75
(*d*) 0.50 (*e*) 0.24.

16. The most popular firing order in the six-cylinder in-line IC engine is
(*a*) 1 — 2 — 3 — 4 — 5 — 6 (*b*) 1 — 3 — 5 — 4 — 6 — 2
(*c*) 1 — 3 — 6 — 5 — 4 — 2 (*d*) 1 — 5 — 3 — 6 — 2 — 4.

17. For same compression ratio and heat input, the cycle which has maximum efficiency may be
(*a*) Diesel cycle (*b*) Dual cycle (*c*) Otto cycle (*d*) None of the above.

18. For the same maximum pressure and heat supplied, the efficiency is maximum for
(*a*) Otto cycle (*b*) Diesel cycle (*c*) Dual cycle (*d*) None of the above.

19. If the working substance in case of air-standard cycle is changed from air to argon for the same compression ratio and heat input at constant volume, the efficiency will
(*a*) decrease (*b*) increase (*c*) remain constant (*d*) none of the above.

20. The pressure at the end of the compression in case of motor car (S.I. engine) is of the order of
(*a*) 7 bar (*b*) 10 bar (*c*) 15.5 bar (*d*) 20 bar.

21. The thermal efficiency of Otto cycle, having same heat input and working substance will
(*a*) increase (*b*) decrease (*c*) remain constant
(*d*) none of the above with increase of compression ratio.

22. In a petrol engine the high voltage for spark is in the order of
(*a*) 1000 V (*b*) 2000 V (*c*) 11 kV (*d*) 22 kV.

23. The material for centre electrode in spark plug is
(*a*) carbon (*b*) platinum (*c*) platinum-tungsten alloy
(*d*) nickel alloy (*e*) none of the above.

24. For economy (minimum fuel consumption), the air-fuel ratio for petrol engine is of the order of
(*a*) 9 : 1 (*b*) 12 : 1 (*c*) 16 : 1 (*d*) 20 : 1.

25. Material for piston in case of petrol engine is
(*a*) cast-iron (*b*) aluminium
(*c*) phosphorus-bronze (*d*) cast steel.

26. The ratings of C.I. engine fuel is given by
(*a*) octane number (*b*) performance number
(*c*) cetane number (*d*) none of the above.

27. The high-vapour pressure fuel of gas turbine is
(*a*) JP-3 (*b*) JP-4 (*c*) JP-5 (*d*) none of the above.

28. The compression ratio of diesel pump engine is in the order of
(*a*) 5 (*b*) 10 (*c*) 16 (*d*) 18.

29. The mechanical efficiency (η_m) of an IC engine is equal to
(*a*) IHP/BHP (*b*) BHP/IHP (*c*) BHP/FHP (*d*) FHP/BHP.

30. The ratio of the indicated thermal efficiency to the corresponding ideal air-standard efficiency is called
(*a*) brake thermal efficiency (*b*) indicated thermal efficiency
(*c*) volumetric efficiency (*d*) relative efficiency.

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136. Tetramethyl lead is a better additive than tetraethyl lead because
(*a*) TML has a lower boiling point (*b*) TEL has a lower boiling point
(*c*) TML has better mixing property (*d*) TEL has a better mixing property.

137. As the engine speed increases it is desirable to
(*a*) advance the ignition timing (*b*) retard the ignition timing.

138. At full throttle operation it is necessary to
(*a*) advance the spark (*b*) retard the spark.

139. The function of a distributor in an S.I. engine is to
(*a*) produce the high voltage for sparking
(*b*) distribute the fuel to the appropriate cylinder
(*c*) allow the exhaust gases to escape from the appropriate cylinder
(*d*) provide the correct firing order in the engine.

140. The pump used for circulating lubricating oil in the engine is
(*a*) of centrifugal type (*b*) of plunger type
(*c*) a gear pump (*d*) any of these.

141. If instead of 4-stroke, we use 2-stroke for the completion of an I.C. engine cycle, there would be a loss of efficiency
(*a*) more in S.I. (*b*) more in C.I. (*c*) equal (*d*) any of these

142. For the same size and weight, a 2-stroke cycle engine would deliver power as compared to that of a 4-stroke
(*a*) about twice (*b*) about 1.7 times (*c*) about 1.9 times (*d*) nearly equal.

143. The purpose of venturi in the carburettor is to work as
(*a*) pump (*b*) compressor (*c*) ejector (*d*) none of these.

144. Vapour lock is caused due to
(*a*) locking carburettor jets due to high vapour pressure
(*b*) excess fuel supply to engine due to faster vaporisation
(*c*) complete or partial stopage of fuel supply due to the vaporisation of fuel in supply system
(*d*) supply of liquid fuel particles to engine.

145. The octane number of compressed natural gas (CNG) is approximately
(*a*) 97 (*b*) 120 (*c*) 87 (*d*) 77.

146. The inlet valve closes after BDC for a low speed engine at
(*a*) 10° (*b*) 30° (*c*) 55° (*d*) 40°.

147. The increase of volumetric efficiency of a C.I. engine will increase
(*a*) B.P. (*b*) brake thermal efficiency
(*c*) bmep (*d*) CO.

148. Morse test can be easily applied to determine I.P. of
(*a*) single cylinder C.I. engine (*b*) multi-cylinder S.I. engine
(*c*) single cylinder S.I. engine (*d*) multi-cylinder C.I. engine.

149. Diesel engines as compared to petrol engines require
(*a*) bigger flywheel (*b*) smaller flywheel
(*c*) same size of flywheel (*d*) no flywheel.

150. The tendency of a petrol engine to knock increases by
(*a*) reducing the spark advance (*b*) scavenging
(*c*) increasing cetane number of fuel (*d*) supercharging
(*e*) both (*c*) and (*d*).

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445. Statement I is false since the performance of an S I engine *cannot* be improved by increasing the compression. Statement II is true, since high octane number tends to suppress detonation, therefore, to some extent fuels of higher octane number will prove useful at higher compression ratio. Thus (*d*) is the correct choice.

446. The formula indicated of power (I P.) involves p_mLAN ; *i.e.,* I P. depends upon mean effective pressure (p_m), length of stroke (L), piston diameter $\left(\text{Area A} = \frac{\pi}{4} D^2\right)$, and speed rotation ($n$). Thus ($c$) is the correct choice

447. $\Delta p \propto v^2$, this relationship is shown in curve (c).

451. **Idling System** compensates dilution of charge ; economiser is used for meeting maximum power range of operation, **acceleration pump** for meeting rapid opening of throttle, and choke for cold starting, thus (b) is the correct choice

454. Because four-stroke engines require heavier flywheels as power stroke comes only once every four strokes and also petrol engine is running at the highest r p.m.

D. Fill in the Blanks

1. Detonation in S I engines is caused by the .... of the charge to burn, while knock in C.I engines is caused by the .... of the charge to burn
2. Of all the three-phases of combustion process in a C I. engine, the .... is the most important.
3. While volatility of the fuel is a determining factor in S.I. engines, the .... of the fuel is the determining factor in C I. engines.
4. Octane number of fuel means the percentage of .... in a mixture of .... and ....
5. .... and .... are reference fuels for measuring octane number of S.I. engine fuels
6. .... and .... are reference fuels for measuring cetane number of C I engine fuels
7. .... is done for increasing the .... efficiency of a diesel engine
8. The quantity of fuel in a .... engine is controlled by the rotation of fuel pump plunger by .... and .... arrangement.
9. The function of a carburettor is to control .... ratio and .... of mixture.
10. Crankcase dilutation is caused if the S I. engine fuels are .... volatile and vapour lock characteristics are caused if the S I engine fuels are .... volatile
11. The Stirling engines are .... combustion engines and would be popular in .... sector
12. Wankel rotary engines are of very .... speed but have some .... problem
13. Engine exhaust emissions can be measured accurately by an .... exhaust gas ....
14. Chemically correct air-fuel ratio is called .... ratio and the ratio of actual mass of air to the theoretical mass of air in a diesel engine is called .... efficiency.
15. Ignition delay of fuel .... as the carbon-hydrogen ratio in the molecules increases.
16. The general formulae for paraffins is ....
17. I C engine .... is designed to remove about 30 per cent of the heat produced in the .... chamber
18. The most commonly used firing order for a six-cylinder four-stroke engine is ....
19. The vibration of the .... induced by a variable torque is called .... vibration.
20. The chemically correct air-fuel ratio is called ....
21. Iso-octane is arbitrarily rated .... octane number.
22. The efficiency of a 4-stroke engine is .... than that of a 2-stroke engine
23. By supercharging the .... and .... of a Diesel engine can be increased.

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Q. 6. *How tetra-ethyl lead (T.E.L.) improves the quality of fuel for S.I. engine ?*

Ans. *Tetra-ethyl lead improves the quality of fuel by delaying auto-ignition and allowing it to occur only at a higher temperature.*

Q. 7. *What do you understand by octane number of 85 and cetane number of 75 ? What is H.U.C.R. ?*

Ans.

- A fuel of octane number of 85, gives the same knock intensity as 85% volume iso-octane plus 15% volume heptane, in a standard similar test.
- The cetane number of a fuel is the percentage by volume of cetane in a mixture of cetane and α-methyl naphthalene ($C_{10}H_7CH_3$) that has the same performance in the standard test engine as that of the fuel. Thus cetane number of 75 means the fuel has the same performance as of mixture of 75% cetane and 25% α-methyl naphthalene, both by volume, in the standard test engine.
- *Highest Useful Compression Ratio (H.U.C.R.).* The tendency of an engine to detonate increases as the compression ratio rises. By further increasing the compression ratio of the engine the detonation will, in time, becomes so severe that the power of the engine will commence to decrease due to overheating. *The compression ratio at which this occurs in a specified test engine, under specified operating conditions, is known as the Highest Useful Compression Ratio or H.U.C.R.*

  This method of classification is now little used as it compares fuels when producing a violent detonation which would not be tolerated in the normal running of any engine.

Q. 8. *Shape of the clearance volume controls the detonation in case of S.I. engine. Comment.*

Ans. Clearance volume has an effect on compression ratio,

since compression ratio $= \dfrac{V_s + V_c}{V_c} = 1 + \dfrac{V_s}{V_c}$.

If clearance volume is reduced, compression ratio is increased which will increase chances of detonation in S.I. engines.

Q. 9. *Discuss the effect of engine variables on ignition lag.*

Ans.

- *Ignition lag is not a period of inactivity but is a chemical process.*
- The ignition lag in terms of *crank angles is 10° to 20° and in terms of seconds 0.0015 seconds or so.*

Effects of engine variables on ignition lag :

(i) Fuel. Ignition lag depends on chemical nature of fuel. *The higher the self ignition temperature of fuel, longer the ignition lag.*

(ii) Mixture ratio. Ignition lag is *smallest for the mixture ratio which gives the maximum temperature.* This mixture ratio is somewhat *richer than the stoichiometric ratio.*

(iii) Initial temperature and pressure. *Ignition lag is reduced if the initial temperature and pressure are increased and the initial temperature and pressure can be increased by increasing the compression ratio.*

(iv) Turbulence. Ignition lag when expressed in degrees of crank rotation *increases linearly with engine speed.* Increasing the engine speed means increasing the turbulence.

Q. 10. *Discuss the effects of the following variables on engine heat transfer :*

(i) Spark advance (ii) Engine output (iii) Pre-ignition and knocking.

Ans. (i) Spark advance. *A spark advance more than the optimum as well as less than the optimum will result in increased heat rejection to the cooling system.* This is mainly due to the fact

Ans. Detonation in S.I. engines and knocking in C.I. engines are fundamentally similar phenomenon. Both are processes of auto-ignition subject to the ignition time-lag characteristics of the fuel-air mixture. The *differences* in the two phenomena are as follows :

1. In the *S.I. engine*, the *detonation occurs near the end of the combustion process,* in the *C.I. engine* detonation *occurs near the beginning of combustion.*

2. In the *S.I. engine*, it is *relatively easy to distinguish between knocking and non-knocking operation as the human ear easily finds distinction.*

3. The *detonation in the S.I. engine is of homogeneous charge causing very high rate of pressure rise and very high maximum pressure. In the C.I. engine the fuel and air are imperfectly mixed and hence the role of pressure rise is normally lower than that in the detonating part of the charge in the S.I. engine.*

4. Since in C.I. engine the fuel is injected into the cylinder only at the end of compression stroke there is *no question of pre-ignition as in the S.I. engine.*

It is most important to care and note that *factors that tend to reduce detonation in S.I. engine increase knocking in the C.I. engine and vice-versa.* The detonation in the S.I. engine is *due to simultaneous auto-ignition of the last part of the charge.* To eliminate detonation in the S.I. engine we want to prevent altogether the auto-ignition of the last part of the average and therefore desire a *long delay period and high self ignition temperature of the fuel.* To eliminate knock in the C.I. engine, we want *auto-ignition as early as possible and therefore desire a short delay period and low self-ignition temperature of the fuel.*

Factors tending to reduce knocking in S.I. and C.I. engines :

Factors	S.I. Engines	C.I. Engines
(*i*) *Compression ratio*	Low	High
(*ii*) *Inlet temperature*	Low	High
(*iii*) *Inlet pressure*	Low	High
(*iv*) *Self-ignition temperature of fuel*	High	Low
(*v*) *Time lag of ignition of fuel*	Long	Short
(*vi*) *r.p.m.*	High	Low
(*vii*) *Combustion chamber wall temperature*	Low	High

Q. 24. *Why does rate of pressure rise during combustion is limited to a certain value ?*

Ans.

- The rate of *pressure rise* is a very important aspect from engine design and operation point of view. It *considerably influences the maximum cycle pressure, the power output and the smooth running of the engine.*
- Higher rate of pressure rise during combustion *cause rough running of the engine because of vibration and jerks produced in the crankshaft rotation.* It also *tends to create a situation conducive to an undesirable occurrence known as knocking.* A higher rate of pressure rise near the end of compression stroke and beginning of power stroke would produce high peak pressure giving increased power output of the engine. But if the rate of pressure rise exceeds about 3 to 3.5 bar per degree of crank rotation, the running of engine becomes rough and noisy. Hence the rate of pressure rise is to be limited to a certain value.

Q. 38. ***What is the difference between ignition timing and firing order ?***

Ans. Ignition timing is *the correct instant for the introduction of spark near the end of compression stroke in the cycle.* The ignition timing is fixed *to obtain maximum power from the engine.*

Firing order is *the order in which various cylinders of a milticylinder engine fire.* The firing order is arranged *to have power impulses equally spaced, and from the point of view of balancing.*

Q. 39. ***State the functions of an ignition coil and a condenser in the battery ignition system of a multi-cylinder S.I. engine.***

Ans. Function of ignition coil and condenser :

(*i*) **Ignition coil.** The *function of the ignition coil is to step up 6 to 12 volts of the battery to a high tension voltage (10000 to 20000 volts) sufficient to promote an electric spark across the electrodes of the spark plugs.* The ignition coil consists of two insulated conducting coils called the primary and secondary windings. The primary winding is connected to the battery, and the secondary winding is connected to spark plugs through the distributor. In order to boost the voltage, the primary winding has a few hundred turns of relatively thick wire, whereas the secondary winding consists of several thousand turns of very fine wire.

(*ii*) **Condenser.** The *function of condenser in the ignition system is to help the rapid collapse of the magnetic field and to store up the energy momentarily when the contact breaker points open, so that due to high voltage it may not jump between the breaker points.*

Without the condenser, the induced current would establish an arc across the contact points when they separate, and therefore the collapse of the field would be prolonged, and the voltage rise in the secondary coil would be slow. Meanwhile most of the energy stored in the magnetic field would be consumed in an arc across the contact breaker points (rather than arc across the spark-plug electrodes).

Q. 40. ***What is the main defference between the battery and electronic systems ?***

Ans. The main difference between the battery and electronic ignition systems is as follows :

- In **battery ignition system** *contact breaker is used for making and breaking the primary circuit of the ignition coil.* This making and breaking of the primary circuit is responsible for providing a high voltage across the spark plug electrodes. The contact breaker consists essentially of a fixed metal point against which another metal point bears. A cam driven by the engine shaft is arranged to open the breaker points whenever an electric discharge is required.
- In **electronic ignition systems** *electronic triggering is used to interrupt a circuit carrying a relatively high current.* It makes an *ideal replacement for the breaker points and the condenser.* Many variations of the electronic ignition system are available.
  — In one of the versions the *contact breaker and the cam assembly of the conventional battery ignition system are replaced by a magneto-pulse generating system which detects the distributor shaft position and sends electrical pulse to an electronic control module. The module switches off the flow of current to the primary coil, inducing a high voltage in the secondary winding, which is distributed to the spark plugs as in the conventional breaker system. The control module contains timing circuit which later closes the primary circuit so that the build up of the primary circuit current can occur for the next cycle.*

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Example 17. *A 4-stroke gas engine develops 3.5 kW B.P. at 160 r.p.m. and at full load. Assuming the following data, find the relative efficiency on I.P. basis and A : F ratio used :*

Volumetric efficiency 87%
Mechanical efficiency 73.5%
Clearance volume ...... 2100 cm³
Swept volume ...... 9000 cm³
Fuel consumption ..... 5 m³/h
Calorific value of fuel ...... 18000 kJ/m³
All working cycles are effective. **(P.U)**

Solution. *Given :* $k = \frac{1}{2}$; B.P. = 3.5 kW ; N = 160 r.p.m. ; $\eta_{vol.}$ = 87% ; $\eta_{mech.}$ = 73.5% ; V_c = 2100 cm³ ; V_s = 9000 cm³ ; gas used = 5m³/h ; C = 18000 kJ/m³.

Relative efficiency, η_{rel} :

Compression ratio, $$r = \frac{V_s + V_c}{V_c} = \frac{9000 + 2100}{2100} = 5.286$$

$$\eta_{air\text{-}standard} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(5.286)^{1.4-1}} = 0.4862 \quad \text{or} \quad 48.62\%$$

$$\text{I.P.} = \frac{\text{B.P.}}{\eta_{mech.}} = \frac{3.5}{0.735} = 4.762 \text{ kW}$$

$$\eta_{th(I)} = \frac{\text{I.P.}}{V_g \times C} = \frac{4.762}{\frac{5}{3600} \times 1800} = 0.19 \quad \text{or} \quad 19\%$$

$$\eta_{relative} = \frac{\eta_{th(I)}}{\eta_{air\text{-}standard}} = \frac{0.19}{0.4862} = 0.39 \quad \text{or} \quad \mathbf{39\%} \quad \textbf{(Ans.)}$$

A / F ratio :

Volume of mixture taken in per stroke

$$= \text{Swept volume} \times \eta_{vol.} \times = 9000 \times 0.87 = 7830 \text{ cm}^3$$

Volume of gas per working stroke

$$= \frac{5}{60} \times \frac{1}{(N/2)} = \frac{5}{60} \times \frac{1}{(160/2)} \times 10^6 \text{ cm}^3 = 1041.7 \text{ cm}^3$$

∴ Volume of air = 7830 − 1041.7 = 6788.3 cm³

∴ A/F ratio by volume $= \frac{6788.3}{1041.7}$ = **6.5 : 1 by volume.** **(Ans.)**

Example 18. *A single-cylinder, four-stroke diesel engine running at 440 r.p.m. with cylinder displacement of 0.006 m³ was arranged to draw air through a calibrated orifice in an air box, the pulsations being sufficiently damped by this procedure. The readings obtained were . barometer 736 mm Hg, air temperature 17°C ; depression in the air box 125 mm of water ; diameter of orifice 25 mm ; co-efficient of discharge 0.62.*

Calculate the volumetric efficiency of the engine referred to inlet conditions.

Solution. *Given* · n = 1 ; N = 440 r.p.m. ; V_s = 0.006 m³ ; barometer reading = 736 mm Hg ; air temperature, T = 17 + 273 = 290 K, diameter of orifice, d_0 = 25 mm = 0.025 m ; C_d = 0.62.

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About the Book

This book on **"Internal Combustion Engines"** has been written to meet exhaustively the requirements of various syllabi in this subject for courses of B.E., B. Tech., B.Sc.(Engg.) of various Indian universities. It is equally suitable for U.P.S.C.(Engg. Services) and Section B – A.M.I.E. (India) Examinations. The book contains 21 chapters in all and an ***"Objective Type Questions Bank"*** at the end.

Salient Features

- The presentation of the subject matter is very systematic and the language of text is in a lucid, direct and easy to understand manner.
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About the Author

Er. R. K. Rajput, born on 15th September, 1944 (coincident with Engineer's Day) is a multi-disciplinary engineer. He obtained his *Master's degree in* ***Mechanical Engineering*** (with Hons. - Gold Medal) from Thapar Institute of Engineering and Technology, Patiala. He is also a *graduate engineer* in ***Electrical Engineering.*** Apart from this he holds memberships of various professional bodies like Member Institution of Engineers (MIE); Member Indian Society of Technical Education (MISTE) and Member Solar Energy Society of India (MSESI). He is also a Chartered Engineer (India). He has served for several years as Principal of "Punjab College of Information Technology", Patiala and "Thapar Polytechnic, Patiala".

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